Heat Engines

selection of the value of a to suit each particular case. With forced draught and a high rate of combustion a may be as low as 60, and the value increases as the draught and rate of combustion diminish.

All the values of a given above are for temperatures on the Fahrenheit scale and Q in B.Th U. For temperatures on the Centigrade scale and Q in C.H.U. the above values of a must be multiplied by $\frac{5}{9}$.

Professor Osborne Reynolds, in a paper read before the Manchester Literary and Philosophical Society in 1874, first called attention to the influence of the velocity of the fluids over the surface of the plate on the rate of heat transmission and suggested the formula $Q = A + B\rho v$, where Q is the amount of heat transmitted per degree difference of temperature of fluid and metal, v is the velocity of flow of the fluid over the surface, ρ is the density of the fluid, and A and B are constants.

Dr. J. T. Nicholson has developed the Osborne Reynolds formula and has given the following general formula[1] for the transmission of heat from a gas flowing along a flue.

$$Q = \left[\frac{\phi}{200} + \frac{1}{40}\sqrt{\phi}\left(1 + \frac{1}{m}\right)\rho v\right](\theta_1 - \theta_2)$$

Q = B.Th.U. transmitted per square foot of flue surface per hour.
θ_1 = temperature of gas flowing along the flue in degrees Fah.
θ_2 = temperature of metal wall of flue in degrees Fah.
$\phi = \frac{1}{2}(\theta_1 + \theta_2)$ = mean gas film temperature in degrees Fah.
ρ = density of gas in lb. per cubic foot.
v = velocity of gas in feet per second.
w = weight of gas flowing per second in lb.
a = area of flue in square inches.
c = perimeter of flue in inches.
$m = a/c$ = hydraulic mean depth of flue or tube.

The following example worked out by Dr. Nicholson will serve to illustrate the application of his formula.

Lancashire boiler flue, 36 inches diameter. Coal burnt on a grate 20 square feet area, 400 lb. per hour. Air 24 lb. per pound of coal. Temperature of gases leaving fire, 2200° F. Temperature of gases at drop flue, 900° F. Steam temperature, 350° F.

Then average value of $\theta_1 = \frac{2200 + 900}{2} = 1550$.

$\phi = \frac{1550 + 350}{2} = 950$, assuming that the temperature of the metal is equal to the temperature of the steam.

Weight of gases per lb. of fuel = 25 lb. $w = \frac{400 \times 25}{60 \times 60}$ lb.

[illegible] $\times 36^2$ square inches. $\rho v = \frac{144 w}{a} = 0{\cdot}4$ lb.

[1] [illegible] heat transmission in steam boilers, as deduced from expe[illegible] *[illegible] the Junior Institution of Engineers*, 1909.

$$m = \frac{\pi \times 36^2}{4 \times 36\pi} \quad \frac{36}{4} = 9 \text{ inches.} \qquad + \frac{1}{m} = 1{\cdot}11.$$

$$\text{Therefore } Q = \left[\frac{950}{200} + \frac{\sqrt{950}}{40} \times 1{\cdot}11 \times 0{\cdot}4\right](1550 - 350)$$

$$= 5{\cdot}092 \times 1200$$

$$= 6100 \text{ B.Th.U. per square foot per hour.}$$

15. Work.—When a force acting on a body causes that body to move, the force is said to do *work*. Also, if a body is moved against a resistance, work is done in overcoming the resistance. The amount of work done depends on the magnitude of the force and also on the distance through which it acts.

In measuring work the unit which is generally used by engineers is the work done when a force of one pound acts through a distance of one foot, this unit being called a *foot-pound*. If the unit taken be the work done when a force of one ton acts through a distance of one foot, it is called a *foot-ton*. The foot-ton is used in measuring large quantities of work. For measuring small quantities of work the *inch-pound*, or the work done when a force of one pound acts through a distance of one inch, is frequently used.

The work done by a force is found by multiplying the magnitude of the force by the distance through which it acts.

16. Turning Moment—Work in Turning.—When a force P acting on a body causes that body to rotate about a fixed axis, the line of action of the force being in a plane perpendicular to that axis, the product of P, the magnitude of the force, and the perpendicular distance R of its line of action from the axis is called the *turning moment* or *torque* of the driving force P. If P is in pounds and R is in feet, the turning moment PR is in *pound-feet* or *foot-pounds;* but if P is in pounds and R is in inches, PR is in *pound-inches* or *inch-pounds*. If the line of action of P is not in a plane perpendicular to the axis of rotation, but makes an angle θ with that plane, then the turning moment is PR $\cos\theta$.

If R, the leverage of P, remains constant during the rotation of the body, and if the magnitude of P is also constant, then if ω is the angle in radians through which the body turns, the distance through which P acts is ωR, and the work done by P is PωR, or Tω, where T is the turning moment. If the leverage R or the force P, or both, should vary, then if T is the *mean turning moment* the work done is Tω.

If the amount of turning is given as n revolutions, then the distance through which P acts is $2\pi Rn$, and the work done is $2\pi PRn$, or $2\pi Tn$.

17. Rate of Work—Horse-power.—The working power of any agent depends on *the amount of work* which it can do in *a given time.* Watt found that a good working horse could do 33,000 foot-pounds of work in one minute, and he introduced this as the unit for measuring the working power of steam engines. A steam-engine or any working agent is said to be of *one horse-power* when it can do 33,000 foot-pounds of work in one minute, or 550 foot-pounds in one second.

Evidently the simple rule for finding the horse-power of any working agent, or the horse-power transmitted by any piece of machinery, is

to divide the number of foot-pounds of work done or transmitted per minute by 33,000, or horse-power equals work per second divided by 550.

Horse-power is a measure of the *rate* of doing or transmitting work.

1 horse-power per hour or 1 horse-power-hour = 33,000 × 60 foot-pounds per hour.

18. Electrical Units and their Mechanical Equivalents.—The electromotive force, or electric pressure of an electric current, is measured in volts, and the strength of the current, or the rate of flow of the electricity across a section of the conductor, is measured in *ampères*. The *power* of a current of 1 ampère at an electrical pressure of 1 volt is called a *watt*. Volts × ampères = watts. 1 horse-power = 746 watts. 1 kilowatt = 1000 watts = 1·3405 horse-power = 44,236 foot-pounds per minute. 1 electrical unit or 1 Board of Trade unit = 1000 watt-hours.

19. Energy.—In mechanics the term *energy* means capacity for doing work.

Potential Energy is energy due to the relative position of one body to another, or of one part of a body to another part, when the two bodies or the parts of the same body are under the action of a force or forces tending to alter their relative positions. For example, a body which is allowed to fall towards the earth may be made to do work; hence before it begins to fall it possesses potential energy, or energy due to its position in relation to the earth. A compressed spiral spring has potential energy, because if it is allowed to resume its unstrained form it can be made to do work. Likewise compressed air possesses potential energy. The energy stored in a piece of coal is potential energy, and under favourable conditions the atoms of the constituents of the coal and the atoms of the oxygen of the air will rush together and produce heat which may be converted into work.

Kinetic Energy is energy due to the motion of a body. A gallon of water at rest at a height of 100 feet above the level of the sea possesses 1000 ft.-lb. of potential energy, and if this water is allowed to fall freely to the level of the sea, without doing work on the way, it will in every position of its fall possess 1000 ft.-lb. of energy, but as it descends its potential energy will diminish, and the remainder of the 1000 ft.-lb. will be stored in the water as kinetic energy. When the gallon of water has fallen 25 feet its potential energy will be reduced to 750 ft.-lb., and its kinetic energy will then be 250 ft.-lb.

If a body of weight W lb. falls freely from rest through a height of h feet it will then have stored in it Wh ft.-lb. of kinetic energy, and its velocity will then be, $v = \sqrt{2gh}$ feet per second. Hence the kinetic energy Wh is equal to $\frac{Wv^2}{2g}$. It is evident that the kinetic energy of a body weighing W lb., and moving with a velocity of v feet per second, will be the same, namely, $\frac{Wv^2}{2g}$, whatever be the cause of the velocity, whether, for example, the cause be the force of gravity, as in a falling body, or the force of an explosion, as in a gun.

20. Kinetic Energy of a Rotating Body.—If an indefinitely small body of weight w lb. be moving with a linear velocity v feet per second in a circle of radius r feet, then its angular velocity ω in radians per second is equal to v/r and its kinetic energy is $\frac{wv^2}{2g} = \frac{w\omega^2 r^2}{2g}$ ft.-lb.

If a body of weight W, rotating about a fixed axis with an angular velocity ω, be divided into indefinitely small parts of weights w_1, w_2, w_3, etc., whose distances from the axis of rotation are r_1, r_2, r_3, etc., respectively, then the kinetic energy of the whole body is

$$\frac{\omega^2}{2g}(w_1 r_1^2 + w_2 r_2^2 + w_3 r_3^2 + \text{etc.}) = \frac{W\omega^2 k^2}{2g} = \frac{I\omega^2}{2g},$$

where k is the radius of gyration, and I the moment of inertia of the body about the axis of rotation. If these expressions give the kinetic energy in ft.-lb., then W must be in lb., k in feet, and I in lb. and foot units.

21. Total Kinetic Energy of a Body.– If a body of weight W rotates about an axis through its centre of gravity with an angular velocity ω, and if the radius of gyration of the body about that axis is k, then its kinetic energy due to its rotary motion is $\frac{W\omega^2 k^2}{2g}$. If the centre of gravity of this body has a linear velocity v, then its kinetic energy due to its motion of translation is $\frac{Wv^2}{2g}$. If the body has both kinds of motion simultaneously, then its total kinetic energy is

$$\frac{W\omega^2 k^2}{2g} + \frac{Wv^2}{2g}$$

22. Mechanical Equivalent of Heat.—Heat and work are mutually convertible the one into the other. In a heat engine the heat produced by the combustion of the fuel used is converted into the work done by the engine. When the brakes are applied to the wheels of a moving train, in order to bring it to rest, the kinetic energy of the train is converted into heat at the rubbing surfaces of the brake blocks and wheels, or if the wheels skid the heat is produced at the rubbing surfaces of the wheels and rails.

Careful experiments have shown that a certain definite number J of foot-pounds of work is equivalent to 1 unit of heat and this number J is called the *mechanical equivalent of heat.* J is 1400 ft.-lb. for 1 C.H.U. or 778 ft.-lb. for 1 B.Th.U.

Joule was the first to make careful and elaborate experiments for the determination of the mechanical equivalent of heat, and it is therefore appropriate that the initial letter of this great scientist's name should be taken to denote its value.

23. Thermodynamics.—That branch of science which deals with the relations between heat and mechanical energy is called *thermodynamics.* There are two main principles or laws of thermodynamics.

The first law of thermodynamics is the equivalence of heat and work as given in the preceding Art.

The second law of thermodynamics may be stated in various ways.

For the present it may be given as expressed by Clausius: It is impossible for a self-acting machine, unaided by any external agency, to transfer heat from one body to another at a higher temperature. Or, more briefly, heat cannot by itself pass from one body to another at a higher temperature.

Exercises I.

1. Express the following temperatures on the Fahrenheit scale in degrees Centigrade, 1800, 600, 203, 72, 25, 0, and − 6.

2. Change the following temperatures on the Centigrade scale into their corresponding temperatures on the Fahrenheit scale, 1500, 550, 102, 20, 8, −16, and −20.

3. Find at what temperature the reading of a thermometer on the Fahrenheit scale will be the same as on the Centigrade scale.

4. Compute the difference between the lengths of two tubes, one being made of brass and the other of steel, at a temperature of 332° F., each tube being 5 feet long at 32° F.

5. If a steel yard measure is correct at 15° C., what will be the error in its length at 40° C.?

6. A wrought iron rectangular tank, whose base is horizontal, contains water to a depth of 35·24 inches when the tank and water are at 15° C. Taking the volumes of 1 lb. of water at 15° C. and 60° C. as 0·0160 c. ft. and 0·0163 c. ft. respectively, and the coefficient of linear expansion of wrought iron as 0·0000117 per degree C., calculate the depth of the water in the tank when the temperature of tank and water is 60° C.

7. Determine the weight of water which a glass flask will hold at 90° C. if it holds 1·35 lb. at 10° C., having given: specific volume of water at 10° C. = 0·0160 c. ft. per lb., specific volume of water at 90° C. = 0·0166 c. ft. per lb., and coefficient of linear expansion of glass = 0·000009 for 1° C.

8. If the mean specific heat of copper between 0° C. and t° C. is given by the expression $0·092 + 0·000015t$, how many units of heat are required to heat 40 lb. of copper from 100° C to 400° C.? Deduce also the specific heat of copper *at* 150° C.

9. A piece of lead weighing 1·5 oz. and having a temperature of 100° C. is dropped into 2·5 oz. of water having a temperature of 13° C. contained by a copper vessel which weighs 2·3 oz. The final temperature of the lead, water, and vessel is 14·4° C. Deduce from this experiment the specific heat of lead, taking the specific heat of copper as 0·092.

10. Assuming that the mean specific heat k of wrought iron between 0° C. and t° C. is given by $k = a + bt$, determine the constants a and b, having given that the mean specific heat is 0·1158 between 20° C. and 100° C. and 0·1558 between 20° C. and 1100° C.

11. A pound of ice at 0° C. was placed in two pounds of water at 75° C. Just when all the ice was melted it was found that the temperature of the mixture was 23·2° C. Determine from this experiment the latent heat of fusion of ice.

12. Taking the specific heat of ice as 0·502 and its latent heat of fusion as 80 C.H.U., calculate the number of units of heat required to convert 2·5 lb. of ice at − 5° C. into water at 15° C.

13. What weight of water at 55° F. must be mixed with 1 lb. of steam at 212° F. so that the resulting temperature may be 95° F.? Take the latent heat of the steam as 970·6 B.Th.U.

14 What weight of steam at 100° C. (latent heat 539·2 C.H.U.) must be mixed with 30 lb. of water at 12° C. in order that the temperature of the mixture may be 45° C.?

15. If 2 lb. of steam at 100° C. be passed into 25 lb. of water at 15° C., compute the temperature of the mixture.

16. Apply Stefan's law to calculate the net amount of heat received per square foot per hour by the walls and roof of a locomotive fire-box, if the temperature of the flame which fills the fire-box is 2140° F. and the temperature of the plates is 380° F.

17. From the results of a boiler trial it was calculated that the heat transmitted from the hot gases to the water was 3500 C.H.U. (6300 B.Th.U.) per square foot of heating surface per hour. The average temperature of the gases was 810° C. (1490° F.) and the temperature of the water was 175° C. (347° F.) The plates were 7–16ths of an inch thick and the coefficient of conductivity was 400 heat units per square foot per hour per degree difference of temperature per 1 inch thickness of plate. Calculate the temperature head for the plate as a percentage of the total temperature head between the hot gases and the water.

18. Using the notation of Fig. 8, p. 14, take $\theta_1 = 840°$ C. (1544° F.), $\theta_2 = 185°$ C. (365° F.), $\delta = 0{\cdot}5$ inch, k for plate $= 400$, thickness of water film $= 0{\cdot}01$ inch, k for water film $= 4{\cdot}1$, thickness of gas film 0·03 inch, k for gas film at 0° C. $= 0{\cdot}13$ and proportional to the $\frac{3}{4}$ power of the mean absolute temperature. Assume mean temperature of gas film to be 520° C. (968° F.). Neglect the difference between t_3 and t_1 and the difference between t_2 and t_4.

Calculate Q, the amount of heat transmitted per square foot of plate per hour, and express the temperature heads for the gas film, the plate, and the water film as percentages of the total temperature head, $\theta_1 - \theta_2$. What would be the value of a in the Rankine formula, $Q = \frac{(\theta_1 - \theta_2)^2}{a}$ in order that it may give the same value of Q?

19. Apply the Nicolson formula given on p. 17, to the tubular heating surface of a locomotive boiler, having given: $\theta_1 = 1550°$ F., $\theta_2 = 350°$ F., $p = 0{\cdot}02$, internal diameter of tubes 1·75 inches. Case I. $v = 50$. Case II. $v = 150$. Also, find for each case the value of a in the Rankine formula in order that it may give the same result.

20. State in foot-pounds the following amounts of energy:—(a) a weight of 40 tons raised 30 feet; (b) a projectile of 40 lb. moving at 2000 feet per second; (c) 3·4 horse-power-hours; (d) 2·5 kilowatt-hours; (e) the calorific energy of one lb. of average coal which is 8370 Centigrade heat units. [B.E.]

CHAPTER II

EXPANSION AND COMPRESSION OF GASES

24. Boyle's Law.—On the results of experiments Robert Boyle (*b.* 1627, *d.* 1691) enunciated the law that *the volume of a given weight of gas varies inversely as the pressure, the temperature remaining the same.* The law may be expressed in symbols as follows. Let V_1 be the volume of a certain weight gas and P_1 its absolute pressure. Let the volume be changed to V_2 and the pressure to P_2, the temperature being unchanged. Then $V_2 : V_1 :: P_1 : P_2$ or $P_1V_1 = P_2V_2$.

Fig. 10 shows how Boyle's law may be represented graphically. Suppose the gas to be enclosed in a cylinder behind a piston. When the piston is in the position shown, the volume V of the enclosed gas is represented by ON, and if NQ, at right angles to ON, be made to represent the pressure P of the gas to any convenient scale, then as the piston is moved out or in the locus of the point Q is a curve AQB which is a rectangular hyperbola of which OX and OY, the axes of volume and pressure, are the asymptotes, and the product PV is constant, the temperature of the gas being constant.

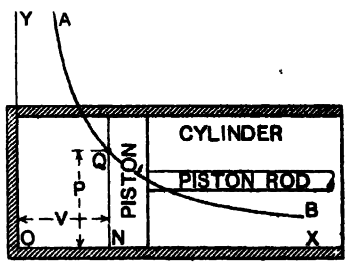


FIG. 10.

Boyle's law is not strictly true for any known gas but, for the gases air, oxygen, nitrogen, and hydrogen the departure from the law may in general be neglected except at very high pressures or very low temperatures.

25. Law of Expansion of Gases by Heat—Absolute Temperature.—The expansion by heat, under constant pressure, of the gases which nearly obey Boyle's law differs from the expansion of solids and liquids in three respects. (1) The coefficients of expansion of these gases are practically constant over a very wide range of temperature, while the coefficients for solids and liquids can only be considered as constant over comparatively small ranges of temperature. (2) The coefficients of expansion of the gases considered are very nearly the same for all, while different solids have generally different coefficients of expansion and so have different liquids. (3) The coefficients of expansion of gases are much greater than those of liquids and still greater than those of solids.

Let V_0 denote the volume of a given weight of gas at 0° C., and let V denote its volume at t° C., the pressure remaining the same, then, $V = V_0(1 + at)$, where a is the coefficient of expansion of the gas

under constant pressure. The coefficient is of course the coefficient of *cubical* expansion. This law of the dilatation of gases by heat was first enunciated by Charles, a French scientist, about 1787, and subsequently in 1802 by Gay-Lussac, another French scientist. The law is called the *law of Charles* or the *law of Gay-Lussac.*

The value of a for hydrogen, which is the gas which most nearly obeys Boyle's law, is 0·003661. For air, $a = 0·003672$ and for nitrogen, $a = 0·003673$.

The value of a for hydrogen may be written $\frac{1}{273·1}$. Taking this value for a, and supposing that the gas is cooled $t°$ below 0° C., the new volume is $V = V_0\left(1 - \frac{t}{273·1}\right)$. As the temperature 273·1° below 0° C. is approached it is evident that V approaches zero volume and therefore approaches a condition in which it can have no heat. This temperature 273·1° below 0° C. or − 273·1° C. is called the *absolute zero* of temperature.

If t is the ordinary temperature on the Centigrade scale, then the absolute temperature is $t + 273·1$.

Referring again to the formula $V = V_0(1 + at)$, this may be written $V = V_0\left(1 + \frac{t}{273·1}\right) = V_0\left(\frac{273·1 + t}{273·1}\right) = V_0\frac{T}{273·1}$, where T is the absolute temperature corresponding to the ordinary temperature t.

The formula $V = V_0\frac{T}{273·1}$ shows that the volume of a given weight of gas, under constant pressure, is proportional to its absolute temperature.

Since 273·1 degrees on the Centigrade scale corresponds to $\frac{9}{5} \times 273·1 = 491·6$ degrees on the Fahrenheit scale, it follows that the absolute zero is 491·6 Fahrenheit degrees below 32° F. Hence the absolute temperature corresponding to the ordinary temperature $t°$ F. is $491·6 - 32 + t = 459·6 + t$.

26. Characteristic Equation of a Gas.—Let V_0 be the volume of a given weight of a gas at the temperature 0° C., or 273·1° absolute, and let its pressure be P_0. Suppose that the volume is changed to V' and the pressure to P, the temperature remaining the same. Then by Boyle's law $PV' = P_0V_0$ and $V' = \frac{P_0V_0}{P}$. Now suppose that the absolute temperature is changed to T, the pressure P remaining the same. Then by the law of Charles the volume V' will change to $V = V'\frac{T}{273·1} = \frac{P_0V_0}{P}\frac{T}{273·1}$. Therefore $PV = \frac{P_0V_0}{273·1}T$. But for a given weight of the gas the product P_0V_0 is a constant, therefore $\frac{P_0V_0}{273·1}$ is a constant and may be denoted by R. Hence $PV = RT$. That is to say, for a given weight of a gas which obeys the laws of the two preceding Arts. the product of the pressure and volume is proportional to the absolute temperature of the gas. The equation $PV = RT$

NOTE.—The number 273·1 given above is generally taken as 273.

is called the *characteristic equation of a gas.* The value of the constant R will depend on the weight of the gas considered, and on the units of pressure and volume, and also on the absolute temperature scale.

Generally, the weight of gas considered is one pound, the pressure is in pounds per square foot, and the volume is in cubic feet. Take the case of air. One pound of air at standard atmospheric pressure, 14·7 lb. per sq. in or 2116·8 lb. per sq. ft., has a volume of 12·39 cubic feet at 0° C. Hence, using the units mentioned above and taking the absolute temperature T in Centigrade degrees,

$$PV = \frac{2116 \cdot 8 \times 12 \cdot 39}{273 \cdot 1}T = 96 \cdot 03T.$$

If the absolute temperature T be in Fahrenheit degrees and the other units are the same as before, then for one pound of air

$$PV = \frac{2116 \cdot 8 \times 12 \cdot 39}{491 \cdot 6}T = 53 \cdot 35T.$$

The characteristic equation PV = RT, in which P, V, and T are variables, is the equation to a surface which is shown in pictorial projection in Fig. 11. OX, OY, and OZ are three axes parallel to which volumes, pressures, and temperatures respectively are measured, and ACBB′C′A′ is the surface referred to.

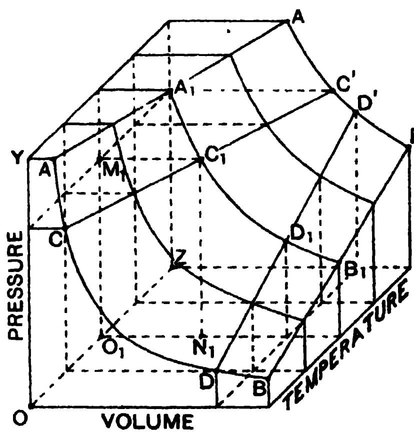


FIG. 11.

For any particular temperature T_1 the product PV is constant and the curve $A_1C_1B_1$ which represents this is a section of the surface by a plane $O_1A_1B_1$ parallel to XOY, the position of this plane being fixed by T_1 which determines the distance OO_1. It is more convenient not to take O on OZ as the absolute zero of temperature but to take it as representing some higher definite temperature θ, then $OO_1 = T_1 - \theta$.

A point C_1 on the surface corresponds to a volume $C_1M_1 = V_1$, a pressure $C_1N_1 = P_1$, and a temperature $OO_1 + \theta = T_1$.

Any section of the surface by a plane parallel to XOY is a rectangular hyperbola, such as $A_1C_1B_1$, the co-ordinates of points in which represent pressure and volume according to Boyle's law.

Any section of the surface by a plane parallel to XOZ is a straight line, such as CC_1C', the co-ordinates of points in which represent volume and temperature according to the law that volume is proportional to absolute temperature when the pressure is constant.

Any section of the surface by a plane parallel to YOZ is a straight line, such as DD_1D', the co-ordinates of points in which are pressure and temperature according to the law that pressure is proportional to absolute temperature when the volume is constant.

27. Internal Energy of a Gas.—When a gas expands and does external work without at the same time receiving heat, the work is done at the expense of the heat or energy in the gas. This energy in the gas by virtue of which it can do work, although it may not all be available for that purpose, is called the *internal or intrinsic energy* of the gas. The total amount of internal energy in a given mass of gas cannot be determined exactly, because its behaviour and its condition as regards the motions of its molecules at extremely low temperatures are unknown. But it is obvious that if a known quantity of heat be given to or taken from a gas, which at the same time does no external work or has no work done on it, its internal energy will be increased or diminished by exactly that amount. In the study of heat engines it is only differences of internal energy which have to be dealt with, so that the total internal energy is not of any importance.

The experiments of Joule showed that *the internal energy of a gas depends on its temperature and is independent of its pressure and volume.* Joule took two equal copper vessels A and B connected by a pipe in which a stop cock was fitted. A was charged with gas at a high pressure (22 atmospheres) and B was exhausted. These vessels were immersed in a tank of water, and on opening the stop cock the gas rushed from A to B until the pressure was the same in both vessels. It was then found that no change of temperature could be detected, and as the net work done by the gas was nothing, the internal energy of the gas was unaltered, notwithstanding that its volume was doubled and its pressure halved.

More delicate experiments were subsequently made by Joule and Thomson (Lord Kelvin) in which a steady current of gas was maintained through a porous plug of cotton wool. The gas on the supply side of the plug was kept at one constant pressure while that on the delivery side was kept at another, but lower constant pressure. In these experiments, as in Joule's earlier ones, no external work was done by the expanding gas, but it was found that with air, oxygen, and nitrogen there was a slight fall in temperature after the throttling process. With hydrogen it was found that there was a slight rise in temperature at ordinary temperatures, but at low temperatures hydrogen behaved like the other gases. This slight cooling while a gas expands without doing work is known as the *Joule-Thomson effect.*

28. Specific Heats of Gases.—When a gas is heated under constant pressure it expands, and external work is done in overcoming the resistance to expansion, and more heat is required to raise the temperature of the gas through a given range than would be necessary if the gas were prevented from expanding. Hence a gas has two specific heats, one for constant pressure and one for constant volume.

Let 1 lb. weight of a gas occupy V cubic feet when at the absolute temperature T and under a pressure of P lb. per square foot. Suppose that heat is given to this gas and that its temperature is raised from T to T_1, the pressure P remaining the same. The volume of the gas will increase to V_1 cubic feet. The external work done in increasing the volume from V to V_1 under the constant pressure P will be $P(V_1 - V)$ ft.-lb. or $AP(V_1 - V)$ heat units, where A is the reciprocal

of J the mechanical equivalent of heat. If k_p is the mean specific heat of the gas at constant pressure between the temperatures T and T_1, then the total heat given to the gas to change its temperature from T to T_1 is $k_p(T_1 - T)$, and the change in its internal energy after heating is $k_p(T_1 - T) - AP(V_1 - V)$.

If the same weight of the same gas is heated at constant volume from the temperature T to the temperature T_1 no external work is done and the change in the internal energy of the gas is therefore $k_v(T_1 - T)$, where k_v is the mean specific heat of the gas between the temperatures T and T_1.

Hence, $k_p(T_1 - T) - AP(V_1 - V) = k_v(T_1 - T)$

and $$AP(V_1 - V) = (k_p - k_v)(T_1 - T)$$

When a perfect gas is heated under constant pressure its volume is proportional to its absolute temperature, then $V_1 = V\frac{T_1}{T}$ and $V_1 - V = V\frac{T_1 - T}{T}$.

Hence $APV\frac{T_1 - T}{T} = (k_p - k_v)(T_1 - T)$, and $PV = J(k_p - k_v)T$.

But for a perfect gas $PV = RT$, therefore $R = J(k_p - k_v)$.

There is now no doubt that the specific heats of gases vary with the temperature, but there is still considerable uncertainty as to the exact values of the specific heats at high temperatures, and even at ordinary temperatures the values as determined by different investigators vary. After a careful examination of the results of the researches of Mallard and de Chatelier, Langen, Schreber, Holborn and Austin, Holborn and Henning, and others, the following formulæ appear to represent fairly well the specific heats of the gases having connection with heat engines. These formulæ are also represented graphically in Fig. 12.

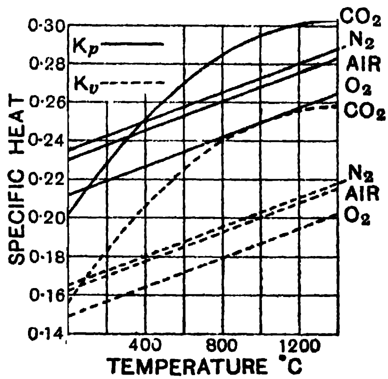


FIG. 12.

Specific Heats of Gases at Constant Pressure.

K_p = specific heat of gas at constant pressure *at* temperature $t°$ C.
k_p = *mean* specific heat of gas at constant pressure from 0° C. to $t°$ C.

Nitrogen (N_2)	$K_p = 0{\cdot}235 + 0{\cdot}000{,}038t.$ $k_p = 0{\cdot}235 + 0{\cdot}000{,}019t.$	Holborn and Henning up to 1400° C.
Oxygen (O_2)	$K_p = 0{\cdot}212 + 0{\cdot}000{,}038t.$ $k_p = 0{\cdot}212 + 0{\cdot}000{,}019t.$	
Air	$K_p = 0{\cdot}230 + 0{\cdot}000{,}038t.$ $k_p = 0{\cdot}230 + 0{\cdot}000{,}019t.$	

According to very careful investigations by Swann, K_p for air is 0·242 at 20° C. and 0·243 at 100° C.

Carbon dioxide (CO_2)—

$$\left\{\begin{array}{l} K_p = 0{\cdot}201 + 0{\cdot}000{,}1484t - 0{\cdot}000{,}000{,}054t^2 \\ k_p = 0{\cdot}201 + 0{\cdot}000{,}0742t - 0{\cdot}000{,}000{,}018t^2 \end{array}\right\} \text{Holborn and Henning up to } 1400° \text{ C.}$$

According to very careful investigations by Swann, K_p for CO_2 is 0·202 at 20° C. and 0·221 at 100° C.

Specific Heats of Gases at Constant Volume.

K_v = specific heat of gas at constant volume *at* temperature $t°$ C.
k_v = *mean* specific heat of gas at constant volume from 0° C. to $t°$ C.

$$\begin{array}{cl} \text{Nitrogen } (N_2) & \left\{\begin{array}{l} K_v = 0{\cdot}165 + 0{\cdot}000{,}038t. \\ k_v = 0{\cdot}165 + 0{\cdot}000{,}019t. \end{array}\right. \\ \text{Oxygen } (O_2) & \left\{\begin{array}{l} K_v = 0{\cdot}149 + 0{\cdot}000{,}038t. \\ k_v = 0{\cdot}149 + 0{\cdot}000{,}019t. \end{array}\right. \\ \text{Air} & \left\{\begin{array}{l} K_v = 0{\cdot}162 + 0{\cdot}000{,}038t. \\ k_v = 0{\cdot}162 + 0{\cdot}000{,}019t. \end{array}\right. \\ \text{Carbon dioxide } (CO_2) & \left\{\begin{array}{l} K_v = 0{\cdot}156 + 0{\cdot}000{,}1484t - 0{\cdot}000{,}000{,}054t^2. \\ k_v = 0{\cdot}156 + 0{\cdot}000{,}0742t - 0{\cdot}000{,}000{,}018t^2. \end{array}\right. \end{array}$$

29. Properties of a Perfect Gas.—Various laws of gases have been stated in the preceding Arts. but it has been pointed out that these laws are only approximately true for actual gases. *A perfect or ideal gas* is one for a given weight of which—(1) Boyle's law is strictly true. (2) The law of Charles is strictly true. (3) $PV = RT$, this being a consequence of (1) and (2). (4) The specific heat at constant pressure is constant, and the specific heat at constant volume is constant. (5) The internal energy or the absolute amount of heat which it contains depends only on its temperature and is therefore independent of its pressure and volume.

For many purposes it may be assumed that the laws of a perfect gas apply to such gases as Hydrogen, Nitrogen, Oxygen, and Air.

30. Isothermal Expansion and Compression.—A gas is said to expand or be compressed *isothermally* when its temperature remains constant during the operation. But when a gas expands or is compressed at constant temperature it may generally be taken as following Boyle's law. Hence the relation between the pressure and volume of a gas which expands or is compressed isothermally is represented by the co-ordinates of the points of a rectangular hyperbola as described in Art. 24 and shown in Fig. 10.

In practice perfect isothermal expansion or compression cannot take place. As a gas expands in a cylinder, doing work on a piston, the gas must lose an amount of heat equivalent to the work done and its temperature must therefore fall, but if the expansion takes place slowly there is time for heat to pass from the cylinder and piston to the gas, the fall in temperature is diminished and an approximation to isothermal expansion is the result, the approximation being closer the slower the rate of expansion.

Also, in compression heat is generated in the gas due to the work of compression, but a portion of this heat is communicated to the cylinder and piston and the rise in temperature of the gas is diminished, and the slower the rate of compression the more nearly will the compression be isothermal.

31. Isothermals for an Imperfect Gas.—The isothermals for hydrogen, nitrogen, oxygen and certain other gases for ordinary temperatures are, for practical purposes, rectangular hyperbolas, but for all gases, as the temperatures and pressures approach those at which the gas assumes the liquid state, the volume decreases more rapidly as the pressure increases than would be the case for a perfect gas.

In illustration of this, various isothermals for carbon dioxide (CO_2) are given in Fig. 13. The upper curve to the right is a rectangular hyperbola and is therefore an isothermal for a perfect gas. The next curve is the isothermal for CO_2 at 48·1° C. and then follow the isothermals for that gas at lower and lower temperatures. Considering the one at 21·5° C., AB is the compression curve while the substance is entirely in the gaseous state. At B the gas begins to liquefy and liquefaction continues at constant pressure, as represented by the horizontal line BC. At C the whole of the gas is liquefied, and if the pressure is continued and increased the volume diminishes very slowly indeed, as shown by the nearly vertical line CD.

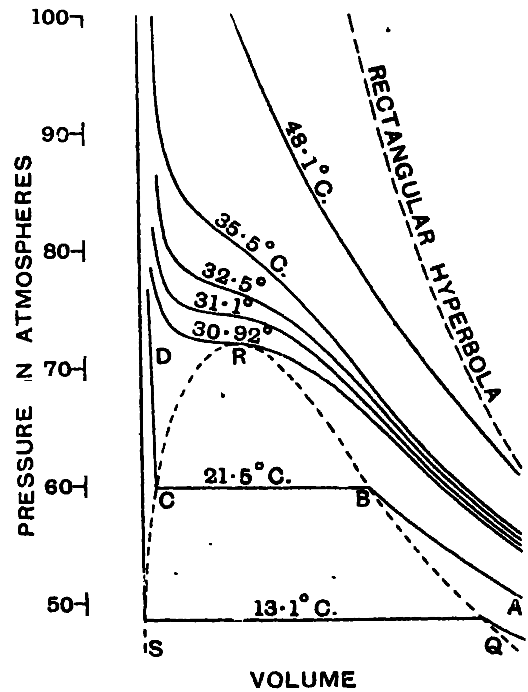


FIG. 13.—Isothermals for CO_2.

If all points such as B be joined and also all points such as C, the dotted curve QBRCS is obtained. Any horizontal intercept, such as BC, of this curve shows the passage of the gas into the liquid state and as the temperature for the isothermals increases the horizontal intercept of the curve QBRCS gets shorter and shorter and ultimately vanishes at the critical point R, where the temperature is about 31° C., which is the critical temperature for CO_2. The corresponding critical pressure is about 73 atmospheres.

Below the critical temperature liquefaction of the gas begins and ends under a constant pressure depending on the temperature, and during this liquefaction the gas or vapour and its liquid exist separately but in contact. No such condition is possible above the critical temperature, and although the gas may be compressed at temperatures above the critical temperature until the density becomes that of the liquid, there is no definite change from gas to liquid at any definite pressure or temperature.

32. Adiabatic Expansion and Compression.—A gas is said to expand or be compressed *adiabatically* when, during the operation, there is no interchange of heat between the gas and any other body. This does not mean that the amount of heat in the gas remains constant. The amount of heat in the gas will diminish if during expansion the gas does work, the diminution in heat being equivalent to the work done. Also, the amount of heat in the gas will increase during compression, the increase being equivalent to the work done in compressing the gas.

In practice perfect adiabatic expansion or compression of a gas cannot take place because of the interchange of heat between the gas and the enveloping cylinder or other vessel containing it, but the more rapidly the expansion or compression is performed the less will be the interchange of heat and the more nearly will the expansion or compression be adiabatic.

33. Energy Equation for a Gas.—If a gas receives a quantity of heat Q while the gas in expanding does an amount of external work W and there is an increase E in the internal energy, then

$$Q = E + W.$$

This is called the *energy equation* for the gas. The quantities of energy Q, E, and W must of course all be measured in terms of the same unit, say the thermal unit or the foot-pound unit.

If the gas is cooled instead of being heated then Q is negative. If the internal energy is diminished then E is negative, and if external work is done on the gas by compressing it, then W is negative.

34. Adiabatic Equation for a Gas: $PV^\gamma = C$.—Consider the case of a unit mass of a perfect gas which is expanding or being compressed. At any instant let the pressure be P in pounds per square foot and the volume V in cubic feet (Fig. 14). Let an indefinitely small change dV take place in the volume V and let the change in temperature be dT.

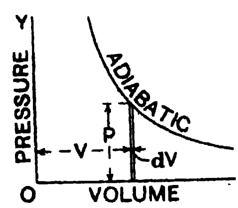


FIG 14.

The external work W done by or on the gas during the indefinitely small change of volume is PdV foot-pounds and the change E in the internal energy is Jk_vdT foot-pounds. Since the change is made adiabatically, Q, the heat added, is nothing, therefore applying the energy equation $Q = E + W$,

$$Jk_vdT + PdV = 0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

But $\frac{PV}{R} = T$. Differentiating this, $\frac{PdV + VdP}{R} = dT$

Substituting this value of dT in (1)

$$Jk_v \frac{PdV + VdP}{R} + PdV = 0. \qquad \text{But } R = J(k_p - k_v)$$

Therefore $$k_v \frac{PdV + VdP}{k_p - k_v} + PdV = 0,$$

which reduces to $$\frac{dP}{P} + \frac{k_p}{k_v} \cdot \frac{dV}{V} = 0 \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Hence, integrating (2), $\log_e P + \frac{k_p}{k_v} \log_e V = \text{constant}$, $\frac{k_p}{k_v}$, the ratio of the two specific heats, is denoted by γ,

Hence $$\log_e P + \gamma \log_e V = \text{constant},$$

and therefore $$PV^\gamma = \text{constant}.$$

When the expansion or compression of a gas does not follow the law $PV^\gamma = \text{constant}$ or the law $PV = \text{constant}$ it generally follows very approximately a law $PV^n = \text{constant}$, where n lies between 1 and 1·4.

35. Geometry of the Curve $PV^n = C$.—The first point to notice with regard to the equation $PV^n = C$ is, that taking logs of both sides, $\log P + n \log V = \log C$, which may be written $y + nx = c$, gives the equation to a straight line. Let AQB (Fig. 15) be the graph of the

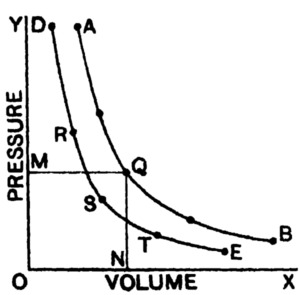


FIG. 15.

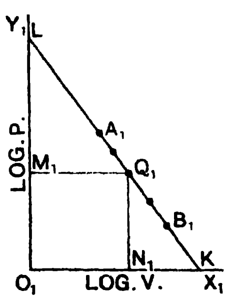


FIG. 16.

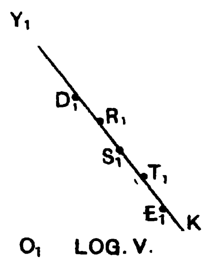


FIG. 17.

equation $PV^n = C$, when $n = 1·4$, and let Q be any point in the curve $QN = P$, and $QM = V$. Take another pair of axes O_1X_1 and O_1Y_1 (Fig. 16). Plot the point Q_1 so that $Q_1N_1 = \log P$ and $Q_1M_1 = \log V$. If this be repeated for other points in the curve it will be found that the points obtained all lie in a straight line LK as shown and $\frac{O_1L}{O_1K} = n = 1·4$. Also $O_1L = \log C$.

Again, let DSE (Fig. 15) be a given curve: it is required to find whether its co ordinates obey the law $PV^n = C$, and if not what is the best value of n which will make $PV^n = C$ approximately true for this curve. A number of points are taken in the curve DSE and the log P and log V for each are found and plotted as in Fig. 17. The points

obtained in this way do not lie exactly in a straight line, which shows that the equation for the curve DSE is not exactly of the form $PV^n = C$. LK is the straight line which most nearly contains all the points, and the best value of n which will make $PV^n = C$ approximately true for the curve is equal to $\frac{O_1L}{O_1K} = 1 \cdot 3$, and the best value of C is found from $\log C = O_1L$.

In working these problems it is not necessary to measure P and V on the true pressure and volume scales; any convenient scales may be used, but preferably the same scale should be used for pressure and volume. That is to say, QN and QM (Fig. 15) may be measured with any linear scale. An ordinary centimetre scale with millimetre subdivisions will generally be found very convenient, and if the centimetre be taken as the unit of pressure and unit of volume then the unit for the logs may be 5 or 10 centimetres.

Having given one point ($P = P_1$ and $V = V_1$) on a curve $PV^n = C$, n being known, other points may be found as follows. Log C is first found from $\log P_1 + n \log V_1 = \log C$. Then assume different values for V and calculate the corresponding values of P from the equation $\log P = \log C - n \log V$.

In working problems on the curve $PV^n = C$ the continued reference to a table of logarithms is avoided by the use of a chart devised by the author and first described in *The Mechanical Engineer*, March 3, 1900. The chart, with three problems worked out on it, is shown in Fig. 18 and is constructed as follows.

XOX_1 and YOY_1 are axes at right angles to one another dividing the paper into four fields. Volumes are measured along OX and pressures along OY, preferably with the same scale as already explained. The curve $PV^n = C$ comes into the field XOY, called the P.V. field, as in Fig. 15.

The logs of the pressures are measured along OX_1 and the logs of the volumes along OY_1. The curve in the field X_1OY, called the P log P field, is plotted from a table of logs, heights above OX_1 being pressures and distances along OX_1 the corresponding logs of these pressures. The curve in the field XOY_1, called the V log V field, is also plotted from a table of logs, but if the pressure and volume scales are the same and the log P and log V scales are the same, which is preferable, then the curve in the V log V field will be the same as that in the P log P field.

The lines corresponding to the line LK in Figs. 16 and 17 come into the field X_1OY_1, called the log field.

Problem I.—Given the point Q on a curve whose equation is $PV^{1 \cdot 4} = C$, it is required to construct the curve.

Through Q draw QQ′ parallel to XOX_1 to meet the P log P curve at Q′ and draw QQ″ parallel to YOY_1 to meet the V log V curve at Q″. Determine Q_1 by completing the rectangle $Q'QQ''Q_1$. Take on OY_1 unit distance OK on the log V scale and make OM on OX_1 equal to n, in this case 1·4, on the log P scale. Join KM. Through Q_1 draw $A_1Q_1B_1$ parallel to MK. Any number of points on the curve required may now be determined by the process which is exactly the reverse of that used for finding Q_1 from Q.

Problem II.—Given the curve DSE: it is required to find whether it is of the form $PV^n = C$, and if not, to find the best value of n which will make $PV^n = C$ approximately true for this curve.

Take a number of points on DSE and find the corresponding points in the log field as shown. It will be seen that these points in the log field are not exactly in a straight line and therefore the curve DSE is not exactly of the form $PV^n = C$. The straight line which most

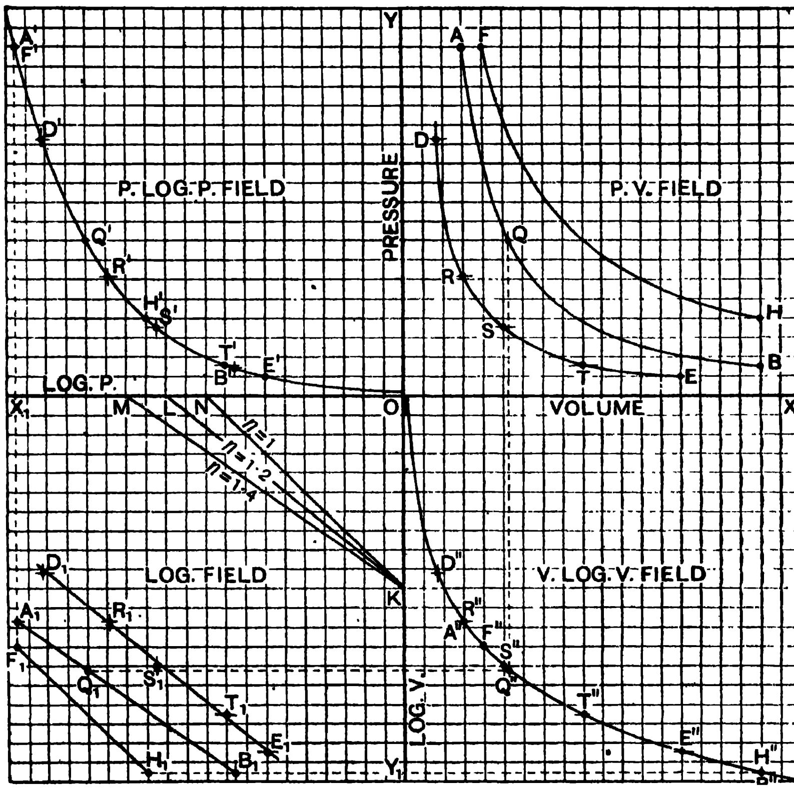


Fig. 18.—Chart for the construction and study of curves of the form $PV^n = C$.

nearly contains these points in the log field is next drawn and a line KL parallel to it determines OL = 1·2, the required value of n.

Problem III.—To draw through the point F the curve FH whose equation is $PV = C$.

In this case $n = 1$. The construction is the same as for the curve AQB and need not be further described.

For the case where $n = 1$ the simple construction shown in Fig. 19 should be used. Q is a given point. To find a point R either to the right or to the left of Q where the volume is ON proceed as follows. Draw QM and NR parallel to OY. Draw QS parallel to OX to meet

NR or NR produced at S. Join OS and produce it if necessary to cut MQ or MQ produced at T. Draw TR parallel to OX to meet NS or NS produced at R. The point R is another point in the curve which is a rectangular hyperbola and is symmetrical about the line OU which bisects the angle XOY.

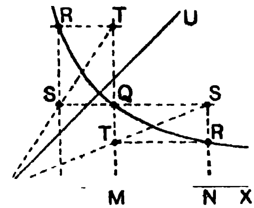


FIG. 19.

Referring again to the chart. The essential parts of the chart are the axes and the P log P and V log V curves and if these are drawn on accurate squared paper, preferably ruled in centimetres and millimetres, points in the different fields may be located without drawing any lines from one field to another. Problems may be worked on a piece of tracing paper placed over the chart. Suppose for example that it is required to examine an expansion or compression curve of an indicator diagram. Make a tracing of the curve together with the axes XOX_1 and YOY_1 which should be known. Placing the tracing over the chart so that the axes on the tracing coincide with the axes on the chart the curve may be dealt with as in Problem II.

Fig. 20 shows various curves of the form $PV^n = C$ in order to illustrate the influence of the index n on the slope of the curve. The

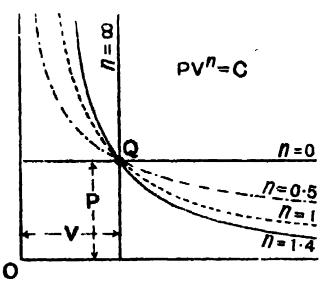


FIG. 20.

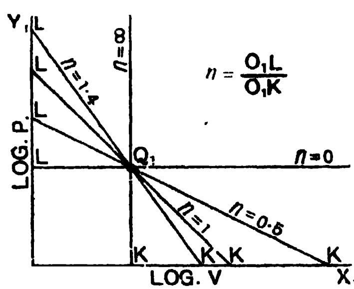


FIG. 21.

curves all pass through the point Q. At any particular pressure or at any particular volume the curve is steeper the larger the value of n. Examining the log P log V diagram (Fig. 21) it is seen that the smaller n is the less is the inclination of LK to O_1X_1, and when $n = 0$, LK which passes through Q_1 becomes parallel to O_1X_1, and log P is constant. Hence P is constant and equal to the pressure at Q. As n increases the inclination of LK to O_1X_1 increases and when n is infinite LK which passes through Q_1 becomes parallel to O_1Y_1 and log V is constant. Hence V is constant and equal to the volume at Q.

The equation $PV^n = C$ may be written $P = CV^{-n}$ Hence $\dfrac{dP}{dV}$

$= -nCV \qquad = -n\dfrac{PV^n}{V^{n+1}} = -n\dfrac{P}{V}$, which gives the slope of the curve at the point where the pressure is P and the volume V. Let Q

(Fig. 22) be this point, and let RT be the tangent to the curve at Q meeting OX at R and OY at T. The slope of the curve at Q = tan θ $= \frac{dP}{dV} = -\frac{QN}{NR}$. Also $\frac{P}{V} = \frac{QN}{QM}$. Hence $-n\frac{QN}{QM} = -\frac{QN}{NR}$.

Therefore $n = \frac{QM}{NR} = \frac{ON}{NR} = \frac{TM}{MO}$.

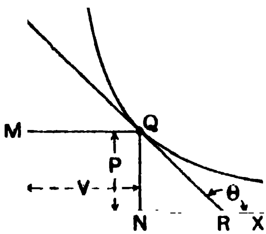


Fig. 22.

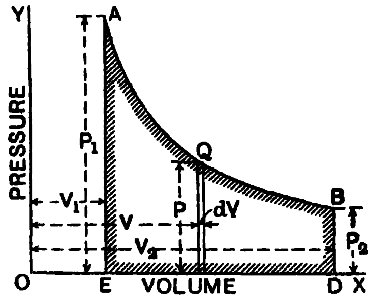


Fig. 23.

36. Work done during Isothermal Expansion or Compression.—Referring to Fig. 23, let AQB be an isothermal expansion or compression curve. This is a rectangular hyperbola whose equation is PV = C. Let the pressures at A and B be P_1 and P_2 and the volumes V_1 and V_2 respectively. From a point Q where the volume is V and pressure P let an indefinitely small change dV take place in the volume V. The work done during this change of volume is $PdV = C\frac{dV}{V}$, and the whole work done during the expansion or compression is

$$\int_{V_1}^{V_2} C\frac{dV}{V} = C \log_e V_2 - C \log_e V_1 = C \log_e \frac{V_2}{V_1} = P_1V_1 \log_e \frac{V_2}{V_1}$$

This work is represented by the area AQBDE.

Using common logs the work done is $2{\cdot}3026\ P_1V_1 \log \frac{V_2}{V_1}$.

If the pressures are in pounds per square foot and the volumes in cubic feet the work is in foot-pounds.

It will be noticed that when V_2 is infinite the total work done during expansion or compression is also infinite, that is, the area under the curve AQB when produced to infinity in the direction OX is infinite.

37. Work done during Expansion or Compression when $PV^n = C$.—Referring to Fig. 23 and proceeding as in the preceding Art., the work done during the change of volume dV is $PdV = C\frac{dV}{V^n}$ and the whole work done during the expansion or compression is

$$\int_{V_1}^{V_2} C\frac{dV}{V^n} = \frac{C}{1-n}V_2^{1-n} - \frac{C}{1-n}V_1^{1-n} = \frac{P_2V_2 - P_1V_1}{1-n} = \frac{P_1V_1 - P_2V_2}{n-1}$$

If the pressures are in pounds per square foot and the volumes in cubic feet the work is in foot-pounds.

It may be pointed out here that V_2 being greater than V_1 then P_2V_2 is less than P_1V_1; for, since $P_2V_2^n = P_1V_1^n$ it follows that $P_2V_2 = P_1V_1\frac{V_1^{n-1}}{V_2^{n-1}}$. In the limit when V_2 is infinite $P_2V_2 = 0$. Also, if $P_2V_2 = RT_2$, then when V_2 is infinite $T_2 = 0$.

The limit to the work done is evidently $\frac{P_1V_1}{n-1}$ when V_2 is infinite.

38. Graphic Representation of the Internal Energy of a Gas.—When a gas expands adiabatically all the external work done is done at the expense of the internal energy of the gas. Hence, referring to Fig. 23, since the work done while the volume changes from V_1 to V_2 is represented by the area AQBDE, then if AQB is an adiabatic expansion curve the area AQBDE will represent the loss of internal energy. Now the theoretical maximum amount of external work which can be got out of the gas is represented by the area under the adiabatic (Fig. 24) when the latter is produced to meet the axis OX at infinity and this is equal to $\frac{P_1V_1}{\gamma - 1}$

The expression $\frac{P_1V_1}{\gamma - 1}$ or its equivalent $\frac{RT_1}{\gamma - 1}$ is frequently quoted as giving the total internal energy of a unit mass of a gas whose pressure is P_1 and volume V_1 or whose absolute temperature is T_1. The assumptions made when this is done are, (1) that the gas behaves

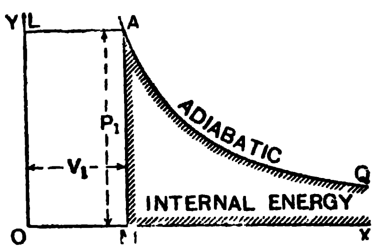


FIG. 24.

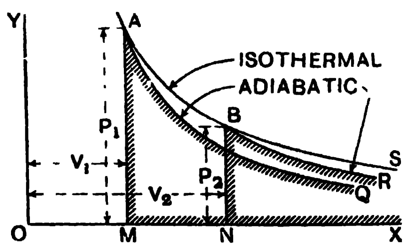


FIG. 25.

as a perfect gas throughout the whole range of the expansion to infinity. and (2) that the gas at absolute zero pressure and temperature has no internal energy. Making these assumptions the total internal energy of a unit mass of gas, in work units (say foot-pounds), is $\frac{RT_1}{\gamma - 1} = Jk_vT_1$, from which, $R = J(k_p - k_v)$, a result which has been already demonstrated in Art. 28.

Referring to Fig. 24 the area of the rectangle ALOM is equal to P_1V_1. Hence the area under the adiabatic AQ from A to infinity in the direction OX is equal to the area of the rectangle ALOM divided by $\gamma - 1$. Taking $\gamma = 1{\cdot}4$ for air the area under the adiabatic from A to infinity is $2{\cdot}5$ times the area of the rectangle ALOM.

If from points A and B in an isothermal ABS adiabatics AQ and BR be drawn, as shown in Fig. 25, the area QAMX, when Q and X are at an infinite distance from O, is $\frac{P_1V_1}{\gamma - 1}$, and the area RBNX, when R and X are at an infinite distance from O, is $\frac{P_2V_2}{\gamma - 1}$. But since A and B are on an isothermal $P_1V_1 = P_2V_2$ therefore $\frac{P_1V_1}{\gamma - 1} = \frac{P_2V_2}{\gamma - 1}$, or the internal energy of the gas as at A is the same as at B. This also follows at once from the fact that the temperature at A is the same as at B.

39. Relations between Pressure, Volume, and Temperature when $PV^n = C$, and $PV = RT$.—Let P_1, V_1, and T_1 be the pressure, volume and absolute temperature of a given weight of gas which expands or is compressed until the pressure is P_2, and let the volume then be V_2 and the absolute temperature T_2.

From the relation $PV^n = C$, $\qquad P_2V_2{}^n = P_1V_1{}^n$

Hence, $\qquad \frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^n \quad$ and $\quad \frac{V_2}{V_1} = \left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}$

From the relation $PV = RT$, $\qquad P_2V_2 = RT_2$ and $P_1V_1 = RT_1$

Hence $\qquad \frac{T_2}{T_1} = \frac{P_2V_2}{P_1V_1} = \left(\frac{V_1}{V_2}\right)^n\left(\frac{V_2}{V_1}\right) = \left(\frac{V_1}{V_2}\right)^{n-1}$

Also, $\qquad \frac{T_2}{T_1} = \frac{P_2}{P_1}\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$

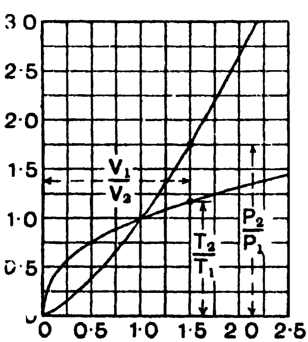


FIG. 26.

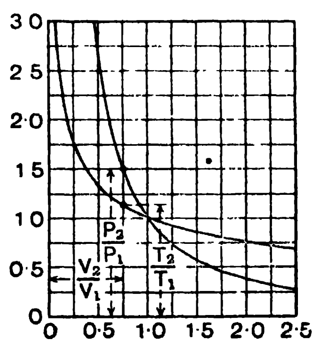


FIG. 27.

The ratios $\frac{P_2}{P_1}$ and $\frac{T_2}{T_1}$ are plotted on a $\frac{V_1}{V_2}$ base in Fig. 26 and on a $\frac{V_2}{V_1}$ base in Fig. 27, for the case where $n = 1{\cdot}4$.

Any one of the above ratios is calculated from either of the other two by means of logarithms as follows:—

$$\log\frac{P_2}{P_1} = n\log\frac{V_1}{V_2} = \frac{n}{n-1}\log\frac{T_2}{T_1}$$

$$\log\frac{V_1}{V_2} = \frac{1}{n}\log\frac{P_2}{P_1} = \frac{1}{n-1}\log\frac{T_2}{T_1}$$

$$\log\frac{T_2}{T_1} = (n-1)\log\frac{V_1}{V_2} = \frac{n-1}{n}\log\frac{P_2}{P_1}$$

40. Heat Received or Rejected by a Gas during Expansion or Compression.—Dealing with one pound weight of the gas, consider it when its pressure is P, volume V, and absolute temperature T, and let an indefinitely small change dV take place in the volume which is accompanied by the indefinitely small change dP in the pressure and the indefinitely small change dT in the temperature, and let dQ be the indefinitely small amount of heat received or rejected.

Stating quantities of energy in thermal units, the change in the internal energy is $k_v dT$, and the external work done is $APdV$, where $A = 1/J$.

But heat received or rejected = change in internal energy + external work done.

$$\text{Therefore } dQ = k_v dT + APdV, \quad \text{and} \quad \frac{dQ}{dV} = k_v\frac{dT}{dV} + AP \quad (1)$$

Note that dQ is positive or negative according as heat is received or rejected by the gas, $k_v dT$ is positive or negative according as the temperature of the gas is raised or lowered, and $APdV$ is positive or negative according as work is done by the gas or done on it.

If $PV = RT$, then

$$\frac{dT}{dV} = \frac{1}{R}\left(P + V\frac{dP}{dV}\right) \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Substituting the value of $\frac{dT}{dV}$ from (2) in (1),

$$\frac{dQ}{dV} = \frac{k_v}{R}\left(P + V\frac{dP}{dV}\right) + AP = \frac{A}{\gamma - 1}\left(P + V\frac{dP}{dV}\right) + AP$$

$$\text{Therefore} \quad \frac{dQ}{dV} = \frac{A}{\gamma - 1}\left(\gamma P + V\frac{dP}{dV}\right) \quad . \quad . \quad . \quad . \quad . \quad (3)$$

If $PV^n = C$, then $V^n\frac{dP}{dV} + nPV^{n-1} = 0$, and $V\frac{dP}{dV} = -nP$

Substituting $-nP$ for $V\frac{dP}{dV}$ in (3), then

$$\frac{dQ}{dV} = \frac{A}{\gamma - 1}(\gamma P - nP) = \frac{\gamma - n}{\gamma - 1}AP \quad . \quad . \quad . \quad (4)$$

$\frac{dQ}{dV}$ is the rate at which heat is received or rejected by the gas

per cubic foot change in volume per pound of gas, and the rate *per second* is obtained by multiplying $\frac{dQ}{dV}$ by the speed of the piston in feet per second at the instant when the pressure is P.

Multiplying both sides of equation (4) by dV,

$$dQ = \frac{\gamma - n}{\gamma - 1} APdV$$

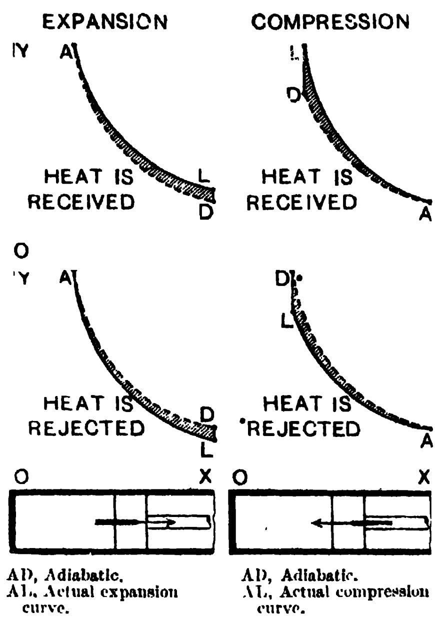
Integrating this between the limits V_1 and V_2, then the heat received or rejected while the volume changes from V_1 to V_2 is

$$Q = \frac{\gamma - n}{\gamma - 1} A \int_{V_1}^{V_2} PdV = \frac{\gamma - n}{\gamma - 1} A \, P_1V_1 - P_2V_2$$

For the case where $n = 1$, $\int_{V_1}^{V_2} PdV = P_1V_1 \log_e \frac{V_2}{V_1}$ and $\frac{\gamma - n}{\gamma - 1} = 1$,

therefore $Q = AP_1V_1 \log_e \frac{V_2}{V_1}$. That is, the whole of the heat received or rejected by the gas is equal to the work done by or on the gas. This also follows from the consideration that when $n = 1$ the expansion or compression is isothermal and there is no change in the internal energy of the gas.

Figs. 28 and 29 show the differences in the expansion or compression curves when heat is received and when heat is rejected by the gas during expansion and during compression. The amount of heat received or rejected per pound of gas is equal to the shaded area, in thermal units, plus the difference between the temperatures at D and L multiplied by k_v, the difference between the temperatures at D and L being taken as positive in each case.



AD, Adiabatic. AL, Actual expansion curve.

FIG. 28.

AD, Adiabatic. AL, Actual compression curve.

FIG. 29.

41. Air Compressors.—The most common form of air compressor for supplying air at high pressure is the cylinder and piston type whose action may be described by reference to its indicator diagram Fig. 30. During the suction stroke a volume of air $AB = V_1$, at pressure P_1, flows into the cylinder through the suction valves which open inwards. During the return or compression stroke this air is compressed and then discharged at a pressure P_2 through the delivery valves which open outwards.

The operation of compression is represented by the curve BD if it is performed isothermally, and by the curve BE if it is performed adiabatically. In practice the compression is neither isothermal nor adiabatic but is intermediate to these conditions of compression and is represented by a curve BF lying between BD and BE.

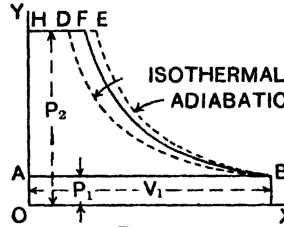


FIG. 30.

When not compressed isothermally the temperature of the air rises during compression and more work is done in compressing and delivering the air. Now when compressed air is used it has almost invariably to be taken some distance from the compressor, and before being used it has assumed the temperature of the surrounding atmosphere. Hence there is no advantage in delivering the air from the compressor at a higher temperature than that of the atmosphere, and there is the disadvantage just mentioned that more work has to be done in the compressor. It is therefore the practice to aim at compressing the air as nearly as possible isothermally by cooling it during compression either by means of a water jacket surrounding the cylinder or by spraying water into the cylinder or by both methods.

The law of compression is $PV^n = C$, where $n = 1$ for isothermal and 1·4 for adiabatic compression. With a water jacket n is commonly 1·3 and with water spray it may be as low as 1·2.

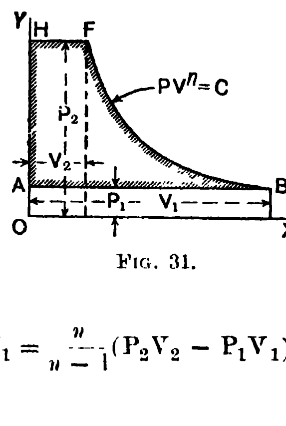


FIG. 31.

The net work done in compressing and delivering a quantity of air whose volume is V_1 at the suction pressure P_1 is evidently represented by the area of the diagram ABFH, Fig. 31. Taking the pressures P_1 and P_2 in lb. per square foot, and the volumes V_1 and V_2 in cubic feet, this work, in ft.-lb. is—

$$W = \frac{P_2V_2 - P_1V_1}{n - 1} + P_2V_2 - P_1V_1 = \frac{n}{n-1}(P_2V_2 - P_1V_1)$$

but $P_2V_2 = P_1V_1\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$

Therefore

$$W = \frac{n}{n-1}\left\{P_1V_1\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - P_1V_1\right\} = \frac{n}{n-1}P_1V_1\left\{\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right\} \quad (1)$$

For isothermal compression $W = P_1V_1 \log_e \frac{P_2}{P_1} + P_2V_2 - P_1V_1$

but $P_2V_2 = P_1V_1$,

therefore $$W = P_1V_1 \log_e \frac{P_2}{P_1} \quad (2)$$

Since isothermal compression is the ideal process, the efficiency of the process assumed in (1) may be expressed by (2) $\div$ (1).

42. Effect of Clearance in Compressors.—In the preceding Art. it has been assumed that the piston works right up to the end of the cylinder and therefore that the whole of the air in the cylinder is expelled. But to safeguard the piston from striking the cylinder end, or cylinder cover, it is necessary to provide a certain amount of clearance. The volume of this clearance space is generally small, about 1 or 2 per cent. of the volume swept through by the piston in one stroke.

The effect of the clearance will be seen by reference to Fig. 32 where HK is the clearance. At the end of the compression and delivery stroke a volume of air represented by HK and having a pressure P_2 is left in the cylinder and when the piston is performing the early part of the suction stroke this air expands until its pressure falls to P_1, its volume then being V_3 represented by LA. The suction valves cannot open until the air in the clearance space has expanded to the pressure P_1. The volume of air taken in during the suction stroke is V_4, represented by AB. The clearance space air of course does work on the piston in expanding and W, the net work done in compressing and delivering the air whose volume is V_4 at the pressure P_1, is represented by the area ABFH.

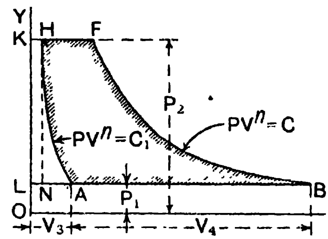


FIG. 32.

Assuming that the value of the index n for the curve HA is the same as that for the curve BF, and making use of equation (1) of the preceding Art.,

$$W = \text{area LBFK} - \text{area LAHK}$$

$$= \frac{n}{n-1} P_1(V_3+V_4)\left\{\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right\} - \frac{n}{n-1} P_1V_3\left\{\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right\}$$

$$= \frac{n}{n-1} P_1V_4\left\{\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right\} \quad .$$

Hence taking V_4 in Fig. 32 the same as V_1 in Fig. 31 it is seen that the net work done in compressing and delivering a given weight of air is independent of the clearance. It is however evident that for a given delivery with clearance the volume swept through by the piston must be greater than the corresponding volume without clearance in the ratio of NB to AB.

43. Volumetric Efficiency.—Referring to Fig. 32, the volume of air taken in during the suction stroke is represented by AB and the stroke volume is represented by NB. The ratio of AB to NB is called the *volumetric efficiency* of the compressor or pump. The *actual volumetric efficiency* is the ratio of the actual volume of air discharged in one delivery stroke, *reduced to standard pressure and temperature*, to the stroke volume. The difference between the apparent and the

actual volumetric efficiencies is due not only to differences in pressure and temperature but also to the leakage from one side of the piston to the other.

44. Multiple-Stage Compressors.—A higher efficiency is obtained by compressing the air in two or more stages in separate cylinders, the air being cooled on its way from the larger to the smaller cylinder. The air can be much more effectively cooled in this way because it can be passed through a long coil of pipe, or through a number of tubes, presenting a large surface to the cooling water.

The advantage of compressing the air in two or more stages is clearly shown in Fig. 33 for two-stage compression, and in Fig. 34 for three-stage compression.

Referring to Fig. 33, AB represents the volume V_1 of air at pressure P_1 taken in by the first or low-pressure cylinder, and BDE is the compression curve assuming that all the compression is performed in the low-pressure cylinder. When the pressure has been raised from P_1 to P_2 the volume has been reduced from AB to MD. The air is then delivered to the second or high-pressure cylinder, but on its way it passes through the inter-cooler as explained above and its volume is reduced to V_2 represented by MK, the point K being on the isothermal BKL. V_2 is the volume of air taken into the high-pressure cylinder. In the high-pressure cylinder the volume is reduced from MK to HF and the pressure is raised from P_2 to P_3. The whole area ABDEH represents the work done in compressing the volume V_1 of air from the pressure P_1 to the pressure P_3 and delivering it at the higher pressure, assuming that this is all done in the low-pressure cylinder, and the shaded area DEFK, compared with the whole area ABDEH, represents the saving due to using two cylinders in succession instead of one.

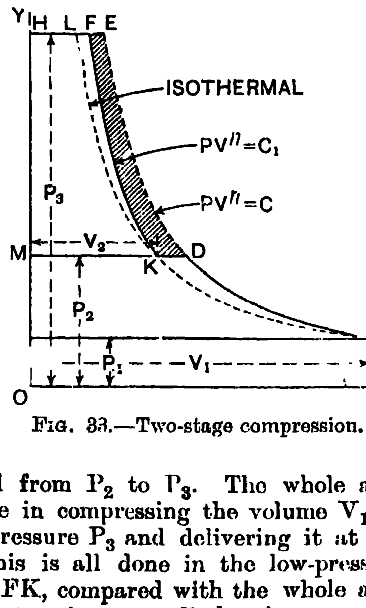


FIG. 33.—Two-stage compression.

The work done in the low-pressure cylinder in compressing the air from volume V_1 and pressure P_1 to the pressure P_2 and delivering it at the higher pressure is represented by the area ABDM and is

$$\text{equal to } \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

The work done in the high-pressure cylinder in compressing the air from volume V_2 and pressure P_2 to the pressure P_3 and delivering it at the higher pressure is represented by the area MKFH and is

$$\text{equal to } \frac{n}{n-1} P_2 V_2 \left[\left(\frac{P_3}{P_2} \right)^{\frac{n-1}{n}} - 1 \right]$$

Let W equal the total work done in the two cylinders, then

$$W = \frac{n}{n-1}P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1}P_2V_2\left[\left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 1\right]$$

But since the point K is on the isothermal BKL, $P_2V_2 = P_1V_1$.

Therefore $W = \frac{n}{n-1}P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} + \left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 2\right]$

If the pressures are in lb. per sq. ft. and the volumes are in cu. ft. then W is in ft.-lb.

The total work done will evidently be a minimum when $\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} + \left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} = y$ is a minimum.

Put $P_2 = x$ and $\frac{n-1}{n} = m$.

Then $y = \frac{x^m}{P_1^m} + \frac{P_3^m}{x^m}$ and $\frac{dy}{dx} = \frac{mx^{m-1}}{P_1^m} - P_3^m m x^{-m-1}$

Therefore y is a minimum when $\frac{x^{m-1}}{P_1^m} = \frac{P_3^m}{x^{m+1}}$

That is when $x^{2m} = P_1^m P_3^m$ or $x^2 = P_1P_3 = P_2^2$ which may also be written $P_2 = \sqrt{P_1P_3}$ or $\frac{P_2}{P_1} = \frac{P_3}{P_2}$

This gives the best value of P_2 when P_1 and P_3 are given.

Since $\frac{P_2}{P_1} = \frac{P_3}{P_2}$ then $\left(\frac{P_2}{P_1}\right)^2 = \frac{P_2}{P_1} \times \frac{P_3}{P_2} = \frac{P_3}{P_1}$

Therefore $\left(\frac{P_3}{P_1}\right)^{\frac{1}{2}} = \frac{P_2}{P_1} = \frac{P_3}{P_2}$

Substituting $\left(\frac{P_3}{P_1}\right)^{\frac{1}{2}}$ for $\frac{P_2}{P_1}$ and for $\frac{P_3}{P_2}$ in the expression for W

$$W = \frac{2n}{n-1}P_1V_1\left[\left(\frac{P_3}{P_1}\right)^{\frac{n-1}{2n}} - 1\right]$$

when the ratios of the pressures are such that the work done is a minimum.

Coming now to three-stage compression (Fig. 34).—The air of volume V_1 at pressure P_1 is compressed to pressure P_2 in the first or low-pressure cylinder and is then delivered through an inter-cooler to the second or intermediate pressure cylinder, its volume shrinking to V_2. The air of volume V_2 at pressure P_2 is compressed to pressure P_3 in the intermediate cylinder, and is then delivered through another inter-cooler to the third or high-pressure cylinder, its volume

shrinking to V_3. The air of volume V_3 and pressure P_3 is compressed to pressure P_4 in the high-pressure cylinder and is then delivered at pressure P_4.

As in the case of two-stage compression the shaded area in Fig. 34 represents the saving due to using three cylinders in succession instead of one.

The total work done is evidently

$$W = \frac{n}{n-1} P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1} P_2V_2\left[\left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 1\right]$$
$$+ \frac{n}{n-1} P_3V_3\left[\left(\frac{P_4}{P_3}\right)^{\frac{n-1}{n}} - 1\right]$$

But $P_1V_1 = P_2V_2 = P_3V_3$

Therefore $W = \frac{n}{n-1} P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} + \left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} + \left(\frac{P_4}{P_3}\right)^{\frac{n-1}{n}} - 3\right]$

W is a minimum when $\frac{P_2}{P_1} = \frac{P_3}{P_2} = \frac{P_4}{P_3} = \left(\frac{P_4}{P_1}\right)^{\frac{1}{3}}$

Substituting $\left(\frac{P_4}{P_1}\right)^{\frac{1}{3}}$ for $\frac{P_2}{P_1}$ and for $\frac{P_3}{P_2}$ and also for $\frac{P_4}{P_3}$

$$W = \frac{3n}{n-1} P_1V_1\left[\left(\frac{P_4}{P_1}\right)^{\frac{n-1}{3n}} - 1\right]$$

The work done per pound of air compressed and delivered is found by substituting RT_1 for P_1V_1 in the foregoing expressions for W.

All the foregoing results are also true when clearance is taken into account provided that V_1, V_2, and V_3 denote the volumes of air taken into the first, second, and third cylinders at pressures P_1, P_2, and P_3 respectively and not the nominal volumes of the cylinders, the nominal volume of a cylinder being the volume swept through by the piston in one stroke.

The work done in compressing and delivering the air in an air compressor as represented by the indicator diagrams of its air cylinder or cylinders may be taken at from 85 per cent. to 90 per cent. of the work required to drive the compressor.

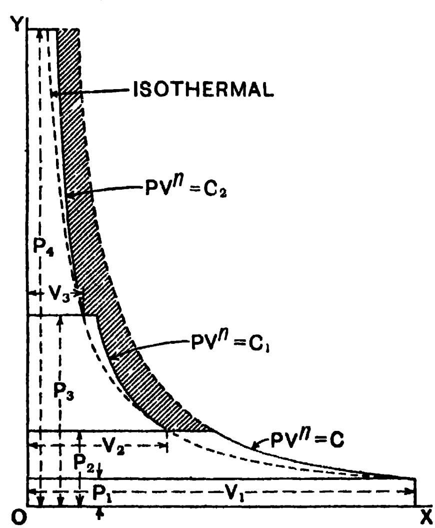


FIG. 34.—Three-stage compression.

45. Compressed Air Motors.—The air to be used in a compressed air motor is conveyed from the compressor reservoir to the motor and there will be a fall in the pressure of the air due to friction in the pipe or main, the fall being greater the greater the distance of the motor from the compressor.

The most common form of compressed air motor is the cylinder and double-acting piston type. The air is admitted into the motor cylinder through a mechanically operated valve and drives the piston forward but after a portion of the stroke of the piston has been performed the air supply is cut off and the stroke is completed under a diminishing pressure as the air expands in the cylinder. After the stroke is completed the air which has done the work is allowed to escape into the atmosphere through a mechanically operated valve. The return stroke is performed by compressed air acting on the other side of the piston. A motor of this type works like a reciprocating steam engine.

In determining the work done in the motor cylinder four cases will be considered. Let P_1 = absolute pressure of the air as it enters the cylinder, and P_2 = absolute pressure of the air as it leaves the cylinder. P_2 will be slightly greater than the pressure of the atmosphere. Also let V_1 = volume of air at pressure P_1 admitted to the cylinder on one side of the piston, and V_2 = volume of air behind the piston when the latter is at the end of its stroke. W = work done by the air whose volume is V_1 at pressure P_1. In the illustrations (Figs. 35 to 37) L denotes the stroke of the piston and also the volume swept through by the piston in one stroke.

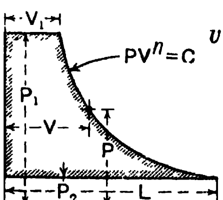


FIG. 35.

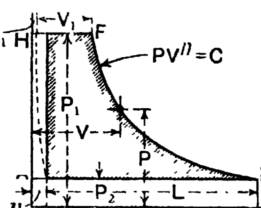


FIG. 36.

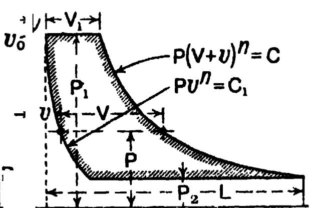


FIG. 37.

CASE I. (Fig. 35).—There is no clearance and the air expands to the exhaust pressure at the end of the stroke.

In this case V_2 = L.

$$W = P_1V_1 + \frac{P_1V_1 - P_2V_2}{n-1} - P_2V_2 = \frac{n}{n-1}(P_1V_1 - P_2V_2)$$

But $P_2V_2 = P_1V_1\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$

Therefore

$$W = \frac{n}{n-1}P_1V_1\left\{1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right\}$$

CASE II. (Fig. 36).—There is a clearance volume v_0 and the air expands to the exhaust pressure P_2 at the end of the stroke.

At the end of the return or exhaust stroke there is left in the cylinder, filling the clearance space, a volume v_0 of air at pressure P_2 and before the piston commences its next stroke the incoming air at pressure P_1 has to raise the air in the clearance space to the pressure P_1. The amount of air required to do this is estimated as follows. Imagine the air filling the clearance space at pressure P_2 to be compressed to the pressure P_1 according to the law $PV^n = C$, and let the volume of this air at the end of the compression be v_1, then

$$P_1 v_1^n = P_2 v_0^n, \text{ from which } v_1 = v_0\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} \text{ and } P_1 v_1 = P_2 v_0\left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}}$$

and the additional volume of air, at pressure P_1, to fill the clearance space is $v_0 - v_1$.

The formula for the area of the diagram ABFH in Fig. 36 is the same as that for the shaded area in Fig. 35 except that V_1 has to be changed to $V_1 + v_1$. Hence,

$$W = \frac{n}{n-1} P_1(V_1 + v_1)\left\{1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right\} - (P_1 - P_2)v_0$$

$$= \frac{n}{n-1} P_1 V_1\left\{1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right\} + \frac{n}{n-1} P_1 v_1\left\{1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right\} - (P_1 - P_2)v_0$$

But $P_1 v_1 = P_2 v_0\left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}}$

Therefore

$$W = \frac{n}{n-1} P_1 V_1\left\{1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right\} + \frac{n}{n-1} P_2 v_0\left\{\left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}} - 1\right\} - (P_1 - P_2)v_0$$

In Case I., $P_2 L^n = P_1 V_1^n$. Therefore $L = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}$

In Case II., $P_2(L + v_0)^n = P_1(V_1 + v_1)^n$

Therefore $L = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}} + v_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}} - v_0$

But $\quad v_1 = v_0\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$

Therefore $L = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}$ as in Case I.

That is to say, for the consumption of the same volume of air V_1 at pressure P_1 the stroke of the piston is the same for Case II. as for Case I.

Case III. (Fig. 37).—There is a clearance volume v_0, the air expands to the exhaust pressure P_2 at the end of the stroke, and the exhaust valve closes before the end of the return stroke, the point at

which it closes being such that the enclosed air at pressure P_2 is compressed to the initial pressure P_1 just as the stroke is completed

$$W = \frac{n}{n-1} P_1(V_1 + v_0)\left\{1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right\} - \frac{n}{n-1} P_1 v_0\left\{1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right\}$$

$$= \frac{n}{n-1} P_1 V_1\left\{1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right\} \text{ the same as in Case I.}$$

The effect of the clearance space is therefore neutralized if there is compression to the initial pressure during the return stroke, but the volume swept through by the piston will have to be greater.

CASE IV. (Fig. 38).—In practice the amount of compression on the return stroke is generally less than that assumed in Case III., also the stroke is shortened so as to cut off the toe of the diagram. The diagram of work is then as shown in Fig. 38. Cutting off the toe of the diagram reduces the size of the cylinder considerably and only a comparatively small amount of work is sacrificed.

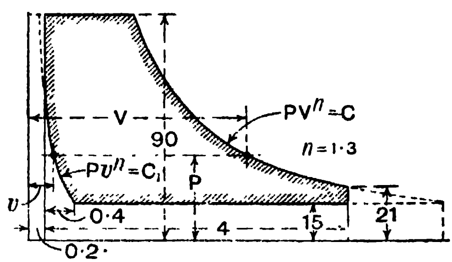


FIG. 38.

Instead of finding the formula for the work done in this case, a formula which would be more complicated than those of the preceding cases, a numerical example will be worked out from first principles.

The data for this example are given in Fig. 38. The volumes are in cubic feet and the pressures in pounds per square inch.

The volume of air behind the piston at cut-off is first calculated and is found to be 1·371 cubic feet. The pressure at the end of the compression is then computed and found to be 62·57 lb. per sq. in

$$W = 144\left\{\frac{90 \times 1{\cdot}371 - 21 \times 4{\cdot}2}{1{\cdot}3 - 1} + 90(1{\cdot}371 - 0{\cdot}2) - 15 \times 4 - \left(\frac{62{\cdot}57 \times 0{\cdot}2 - 15 \times 0{\cdot}6}{1{\cdot}3 - 1} - 15 \times 0{\cdot}4\right)\right.$$

$$= 22{,}605 \text{ foot-pounds.}$$

At the end of the compression, before any additional air is admitted, the clearance space contains 0·2 c. ft. of air at a pressure of 62·57 lb. per sq. in. and if the compression is imagined to be continued according to the same law until the initial pressure of 90 lb. per sq. in. is reached the volume would then be 0·151 c. ft. The volume of air used for the above amount of work is therefore 1·371 − 0·151 = 1·22 c. ft. at 90 lb. per sq. in. pressure.

If the initial temperature of the air is, say, 15° C. then the weight of 1·22 cubic feet of it at 90 lb. per square inch pressure is $1{\cdot}22 \times \frac{90}{14{\cdot}7} \times \frac{273}{273 + 15} \times \frac{1}{12{\cdot}39} = 0{\cdot}57$ lb., where 12·39 is the volume

in cubic feet of 1 lb. of air at standard atmospheric pressure and temperature 0° C.

46. Fall in Temperature During Expansion—Advantage of Preheating the Air.—If P_1 and P_2 are the pressures of the air at the beginning and end of expansion in a compressed air motor, and if T_1 and T_2 are the corresponding absolute temperatures, then $\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$ where n is the index in the equation to the expansion curve.

The air arrives at the motor from the compressor at atmospheric temperature and the fall in temperature is generally very considerable. For example,—Let P_1 = 90 lb. per sq. in., and P_2 = 15 lb. per sq. in., and let the atmospheric temperature be 15° C. (59° F.). Taking $n = 1{\cdot}3$,

$\frac{T_2}{273 + 15} = \left(\frac{15}{90}\right)^{\frac{1{\cdot}3-1}{1{\cdot}3}}$. Therefore $T_2 = 288 \times \left(\frac{1}{6}\right)^{\frac{3}{13}}$. From which $T_2 = 190{\cdot}5$ and $t_2 = 190{\cdot}5 - 273 = -82{\cdot}5°$ C. ($-116{\cdot}5°$ F.).

When the temperature of the expanding air falls below the freezing point the moisture in it is frozen and the snow formed may seriously interfere with the working of the exhaust valve.

The most satisfactory way of preventing too low a temperature at the end of the expansion is to preheat the air at constant pressure by passing it through a suitable heater. By preheating the air not only is the freezing effect avoided but the volume of the air is increased, the volume being proportional to the absolute temperature, and consequently a large proportion of the heat expended in the heater is converted into work in the motor cylinder. If W_1 is the amount of work done in the motor cylinder for the consumption of a given weight of air at the initial pressure, without preheating, its absolute temperature being T_1, and if W_2 is the work done for the same weight of air at the same pressure after it has been heated to the absolute temperature T_2, then, assuming that the work done is proportional to the volume of air used,

$$W_2 = W_1 \frac{T_2}{T_1}$$

Exercises II

Where pressure is mentioned without qualification "absolute" pressure is to be understood.

1. A vessel contains 50 cubic feet of air at a pressure of 100 lb. per sq. in. If one-fifth of the air be removed by an air pump what will be the pressure of the remaining air, the temperature being unaltered?

2. An air receiver has a volume of 15 cubic feet and contains air at a pressure of 80 lb. per sq. in. and at a temperature 15° C. Taking the specific volume of air as 12·39 cubic feet per pound at 14·7 lb. per sq. in. pressure and temperature 0° C., find the pressure in the receiver at 15° C. after 2 lb. weight of additional air has been pumped into it. Assume that the volume of air at constant pressure is proportional to its absolute temperature.

3. A certain weight of air at 20° C. and 40 lb. per sq. in. pressure has a volume of 15 cubic feet. What is the volume of this air at 0° C. and at 70° C., the pressure remaining the same? Also what is the weight of this air if 1 lb. of air at 0° C. and 14·7 lb. per sq. in. pressure is 12·39 cubic feet? Coefficient of expansion of air = 0·003672 per ° C.

4. Air at 14·7 lb. per sq. in. pressure and having a volume of 25 cubic feet is compressed at the constant temperature of 15° C. until its volume is 5 cubic feet. What is then its pressure? The air is then heated until its temperature is 55° C. What is then its volume? Coefficient of expansion of air = 0·003672 per ° C.

5. Find the value of R in the characteristic equation PV = RT for hydrogen, having given that one pound weight of hydrogen has a volume of 178·57 cubic feet at 0° C. and 14·7 lb. per sq. in. pressure. P to be in lb. per sq. ft., V in cubic feet, and T absolute temperature in Centigrade degrees.

6. The volume, pressure, and temperature of 1 lb. of air are, 5 cubic feet, 50 lb. per square inch, and 102° C. respectively. Determine the temperature and pressure of this air after 126 C.H.U. of heat have been given to it at constant volume. Take the mean specific heat of air at constant volume as 0·18.

7. A quantity of gas occupying 3·2 cubic feet, and exerting a pressure of 14 lb. per square inch, and having a temperature of 20° C., is compressed until its volume is 2 cubic feet. If its pressure is then 25·5 lb. per square inch what is its temperature? If the law of compression is $PV^n = C$ what is the value of n?

8. Taking the specific heat of air at constant pressure as 0·237 and at constant volume as 0·168, calculate the volume of 5 lb. weight of air at a pressure of 50 lb. per square inch and at a temperature of 25° C.

9. Write down the characteristic equation for a gas and apply it to solve the following problems:—One pound of air at a pressure of 2 atmospheres has its temperature raised from 60° F. to 600° F. (*a*) at constant pressure; (*b*) at constant volume; determine, for each case, the final pressure and volume, the amount of heat supplied, the change of internal energy, and the external work done. $k_p = 0·238$, $k_v = 0·169$. [U.L.]

10. Find the difference between the work done in compressing 5 cubic feet of air at a pressure of 15 lb. per square inch to a volume of 1 cubic foot adiabatically and isothermally.

11. A gas is compressed from a volume of 5 cubic feet to a volume of 2 cubic feet according to the law $PV^{1·35} = C$. The initial pressure and temperature of the gas are, 13·5 lb. per square inch and 130° C. Calculate the final pressure and temperature and the work done.

12. If 1 lb. of air is expanded from an initial pressure of 200 lb. per square inch absolute, and temperature 1100° C. (2012° F.) to a final pressure of 40 lb. per square inch absolute, (*a*) adiabatically, (*b*) by a process defined by the equation $PV^{1·2}$ = constant, calculate the final temperature, and the work done in each case. [U.L.]

13. The pressure and volume for one point on an expansion curve are, 150 lb. per square inch and 1·3 cubic feet. The pressure and volume for another point are, 35 lb per square inch and 4·25 cubic feet. Assuming the law of the expansion to be $PV^n = C$, find n.

14. The co-ordinates P and V of four points in the expansion curve of an indicator diagram were measured in centimetres and are here tabulated. Find the value of n in the formula $PV^n = C$ which most nearly agrees with this curve.

P	4·05	2·45	1·45	1·0
V	2·0	3·0	4·5	6·0

15. One pound of air at 354° F. (178·9° C.) expands adiabatically to three times its original volume, and in the process falls in temperature to 60° F. (15·6° C.). The work done during the expansion is 88,410 foot-pounds. Calculate the two specific heats. [U.L.]

16. Air at a temperature of 59° F. (15° C.) is compressed in a cylinder from 15 lb. pressure absolute to 120 lb. pressure absolute per sq. inch. The equation of the compression curve is $PV^{1·25} = C$. Find the work done in compressing a pound of air, and the heat that escapes through the cylinder walls [U.L.]

17. In a trial of an air compressor the absolute pressure and volume at two points A and B were as follows:—At A, pressure 20 lb per sq. in., volume 10·6 cu. ft. At B, pressure 40 lb. per sq. in., volume 6·22 cu. ft. Find the heat rejected to the walls between A and B if the weight of air compressed was one pound. Take $k_p = 0·238$ and $k_v = 0·169$. [U.L.]

18. Find the horse-power required to drive an air compressor which has to

compress 1200 cubic feet of air per minute from 15 lb. per square inch to 60 lb. per square inch and deliver it at the higher pressure. Assume that the index n for the compression curve is 1·25 and that 12 per cent. of the work supplied to the compressor is wasted.

19. An air receiver having a capacity of 50·8 cubic feet has to be filled with air at a pressure of 80 lb. per square inch. At the beginning of the operation the receiver contains air at atmospheric pressure, 14·7 lb. per square inch. Find the work to be done assuming that the law of compression is $PV^{1.3} = C$.

20. In designing an air compressor from certain data it was calculated that without clearance the stroke of the piston should be 25 inches. The compressor was required to compress the air from 15 lb. per square inch to 75 lb. per square inch. It was decided to add a linear clearance of 3-8ths of an inch. Determine the proper length of stroke so that the delivery per stroke with this clearance may be the same as that without clearance. Assume the index n in the expansion curve to be 1·25.

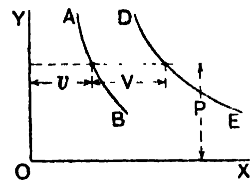


FIG. 39.

21. AB (Fig. 39) is a curve whose equation is $Pv^n = C_1$. DE is another curve whose equation is $P(V + v)^n = C_2$. The value of n is the same for both curves. Show that $PV^n = C_3$, and that $v/V = C_4$, where C_1, C_2, C_3, and C_4 are constants. Find also the values of C_3 and C_4 in terms of C_1, C_2, and n.

22. A single-stage double-acting air compressor has a cylinder 20 inches in diameter and a piston stroke of 12 inches. The clearance volume at each end of the cylinder is 2 per cent. of the volume swept through by the piston in one stroke. The suction pressure is 14 lb. per square inch, the delivery pressure is 45 lb. per square inch, and the law of compression and expansion is $PV^{1.3} = C$. The atmospheric pressure is 15 lb. per square inch, and the piston makes 500 strokes per minute. Determine the number of cubic feet of "free air," that is, air at atmospheric pressure, compressed and delivered per minute. Find also the horse-power to drive this compressor if the mechanical efficiency is 87 per cent.

23. The L.P. cylinder of a compound air compressor draws in 4 c. ft. of air at a temperature of 59° F. (15° C.) and at a pressure of 15 lb. per sq. in. ab. It compresses this adiabatically to 30 lb. per sq. in. ab., and then delivers it into a receiver where the air is cooled under constant pressure to 77° F. (25° C.). This air is then drawn into the H.P. cylinder and compressed adiabatically to 80 lb. per sq. in. ab. and delivered to the reservoir. Find the indicated horse-power when the compressor works at 100 revs. per minute, double acting. What pressure in the receiver would give the best efficiency, assuming the other data as given above? [U.L.]

24. A double-acting two-stage compressor has a high-pressure cylinder 18 inches in diameter and a low-pressure cylinder 30 inches in diameter, the common stroke being 42 inches. During a test 380 cubic feet of air at a pressure of 80 lb. per square inch absolute and temperature 78° F. (25·56° C.) are discharged from the H.P. cylinder per minute. The mean effective pressures in the H.P. and L.P. cylinders are 34 lb. per square inch and 13·5 lb. per square inch respectively, and the revolutions are 75 per minute. The atmospheric temperature is 62° F. (16·67° C.) and the barometer is at 29·8 inches. Calculate the efficiency of the compression—that is, the ratio of the work required for isothermal compression, to the work actually done on the air. [U.L.]

25. For each stroke of the piston of a compressed-air motor one cubic foot of air at 90 lb. per square inch pressure is taken into the cylinder, which is 20 inches in diameter, and expanded to the exhaust pressure which is 15 lb. per square inch. Determine the stroke of the piston, (*a*) when there is no clearance (Fig. 35, p. 45), (*b*) when there is a clearance volume of 5 per cent. of the volume swept through by the piston in one stroke, and there is compression up to the initial pressure during the return stroke (Fig. 37, p. 46). In each case take $n = 1.3$.

26. The air supplied to a compressed-air motor has a pressure of 84 lb. per sq. in., and a temperature of 20° C. (68° F.). In the motor cylinder the air expands to the exhaust pressure, 15 lb. per sq. in., at the end of the stroke. The piston, which is double acting, has a diameter of 10 inches and a stroke of 12 inches, and

there are 500 strokes per minute. The clearance volume is 5 per cent. of the stroke volume, and there is compression up to the initial pressure. Calculate: (*a*) The weight of air used per minute. (*b*) The horse-power developed in the cylinder. (*c*) The weight of air used per horse-power per hour. (*d*) The temperature of the air at the end of the expansion. Take $n = 1{\cdot}3$, and the volume of 1 lb. of air at 14·7 lb. per sq. in. pressure and 0° C. (32° F.) temperature as 12·4 cubic feet.

27. The initial pressure of the air in a compressed-air motor is 75 lb. per sq. in. and the final pressure is 15 lb. per sq. in. To what temperature must the air be preheated in order that the temperature after expansion may be 2° C. (35·6° F.)? Take $n = 1{\cdot}3$.

CHAPTER III

PROPERTIES OF STEAM

47. Pressure Energy of a Liquid.—AB (Fig. 40) represents a vertical cylinder fitted with a piston of area a square feet. On the piston there is a constant total load W pounds which includes the weight of the piston and the pressure of any air, steam, or gas above the piston. The piston is raised by pumping a liquid into the cylinder beneath the piston.

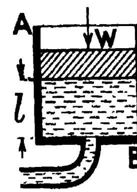


FIG. 40.

Let p denote the pressure of the liquid in pounds per square inch, and let the piston be raised a distance l feet. Evidently $144\ pa = W$, and the work done is $Wl = 144\ pal$ foot-pounds. The volume of the liquid introduced into the cylinder is $al = u$ cubic feet. Therefore $144\ pu =$ work done. If u is the volume of one pound of the liquid then $144\ pu$ is the work done in pumping one pound of the liquid into the cylinder under the above conditions. If the operation is reversed W will force the liquid out of the cylinder and the liquid as it is discharged may be made to do work amounting to $144\ pu$ foot-pounds per pound of liquid. Hence a pound of liquid under a pressure of p pounds per square inch and having a volume u cubic feet possesses $144\ pu$ foot-pounds of energy due to its pressure and this is called the *pressure energy* of the liquid.

48. Formation of Steam in a Boiler.—Imagine a steam boiler, shown at (a) Fig. 41, to contain a cylinder AB fitted with a piston.

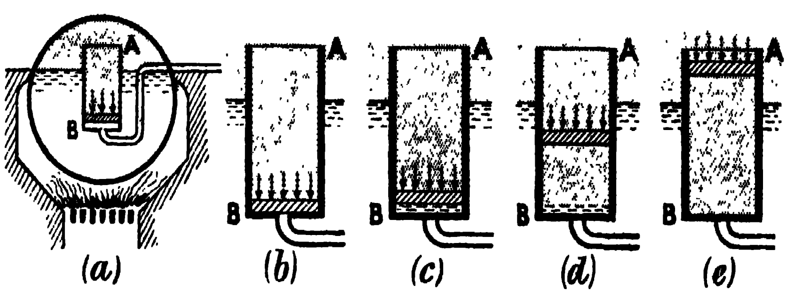


FIG. 41.

The cylinder is immersed in the water in the boiler and the lower end of the cylinder is closed except for an opening into a feed pipe which passes to the outside of the boiler. The space in the cylinder above the piston is in free communication with the steam space of the boiler.

Let the steam in the boiler have a constant absolute pressure p pounds per square inch and temperature $t°$. At (b), (c), (d), and (e)

the cylinder is shown separately to a larger scale. As shown at (*b*) the piston is at the bottom of the cylinder. One pound of water, to be called the feed water, at 0°C (32°F.) and having a volume u cubic feet is now pumped into the cylinder underneath the piston and the opening into the cylinder from the feed pipe is then supposed to be closed. The conditions are then as shown at (*c*). The external work required to force the one pound of feed water into the cylinder is $144\ pu$ foot-pounds and the equivalent of this in heat units is $144\ Apu$ where A is the reciprocal of the mechanical equivalent of heat.

Heat passes from the water outside the cylinder to the water within it and the temperature of the latter is raised to $t°$, the temperature of the water and steam in the boiler. The amount of this heat is denoted by h. The heat h added to the feed water to raise its temperature from 0° C. (32° F.) to $t°$ causes it to expand slightly, its volume increasing from u to u', and the piston is therefore slightly raised and further external work is done amounting to $144Ap(u'-u)$ heat units. It will shortly be shown that this is a negligible quantity. Up to this point the total external work done is therefore $144Apu'$ heat units.

The flow of heat continuing through the cylinder to the water within it, the latter begins to boil and evaporate, steam is formed and the piston is raised. As shown at (*d*) part only of the feed water in the cylinder has evaporated. Ultimately the whole of the one pound of feed water in the cylinder is evaporated and the cylinder then contains v cubic feet of steam below the piston as shown at (*e*). From the instant that the water beneath the piston begins to evaporate to the instant that it is all changed into steam the volume beneath the piston increases from u' to v and the external work to do this is $144\ Ap(v - u')$ heat units.

If L is the amount of heat required to convert the one pound of feed water at the boiling temperature $t°$ into steam at the same temperature, then the whole heat required to convert one pound of water at 0° C. (32° F.) into steam at $t°$ is $h + L = H$. The part h is called the *sensible heat* or *heat of the liquid* and L is the *latent heat of the steam.* The quantity H has been generally called the *total heat of the steam* but H does not however represent the whole of the energy required to produce one pound of steam in a boiler from water at the freezing temperature. There is in addition the pressure energy to be given to the feed water to get it into the boiler, which has been shown to equal $144Apu$ heat units.

From this consideration the total energy of the steam is $H' = H + 144Apu$. Similarly the total energy of the water is $h' = h + 144Apu$. Various values of $144Apu$ are given in the table below and from these it will be seen that for practical purposes the term $144Apu$ is negligible when it is compared with h or H as found from steam tables.

p (lb. per sq. in. ab.)		15	115	215	315
$144\ Apu$	C.H.U.	0·03	0·21	0·41	0·61
	B.Th.U.	0·05	0·38	0·74	1·10

It may be observed that the total external work done in the production of one pound of steam in a boiler is

$$144Apu + 144Ap(u' - u) + 144Ap(v - u') = 144Apv$$

The *internal* or *intrinsic energy* of steam is its total energy minus the external work done in producing it.

$$\text{Internal energy} = h' + L - 144Apv$$

Having studied the formation of steam from water introduced under a piston in a cylinder inside a boiler, in the manner described, no difficulty should be experienced in seeing that the results are exactly the same when the cylinder and piston are dispensed with and the feed water is pumped directly into the boiler as in ordinary practice.

It will be instructive to reconsider the foregoing imaginary experiment, taking numerical values. Suppose the steam in the boiler to exert an absolute pressure of 200 lb. per sq. in. The corresponding temperature is 194·4° C. $H = 667{\cdot}7$ C.H.U. $L = 470{\cdot}3$ C.H.U. $h = H - L = 197{\cdot}4$ C.H.U. $u = 0{\cdot}0160$ c. ft. $u' = 0{\cdot}0184$ c. ft. $v = 2{\cdot}30$ c. ft.

Pressure energy of 1 lb. of water at 0° C. fed into boiler

$$= 144 \times 200 \times 0{\cdot}016 = 460{\cdot}8 \text{ ft.-lb. or } \frac{460{\cdot}8}{1400} = 0{\cdot}329 \text{ C.H.U.}$$

External work done in expanding water

$$= 144 \times 200(0{\cdot}0184 - 0{\cdot}0160) = 69{\cdot}12 \text{ ft.-lb. or } \frac{69{\cdot}12}{1400} = 0{\cdot}049 \text{ C.H.U.}$$

Total energy of water at boiling temperature

$$= 197{\cdot}4 + 0{\cdot}329 + 0{\cdot}049 = 197{\cdot}778 \text{ C.H.U.} = h'$$

External work done in evaporating the water

$$= 144 \times 200(2{\cdot}30 - 0{\cdot}0184) = 65710{\cdot}08 \text{ ft.-lb.}$$

or

$$\frac{65710{\cdot}08}{1400} = 46{\cdot}936 \text{ C.H.U.}$$

Total external work done in producing steam

$$= 46{\cdot}936 + 0{\cdot}049 + 0{\cdot}329 = 47{\cdot}314 \text{ C.H.U.},$$

which is also equal to $144 \times 200 \times 2{\cdot}3 \div 1400$.

Total energy of steam

$$= 667{\cdot}7 + 0{\cdot}329 + 0{\cdot}049 = 668{\cdot}078 \text{ C.H.U.} = H'$$

In this work the pressure energy of the water, $144Apu$, *and the external work done in expanding the water, will in general be neglected.* This means that the difference between H' and H and the difference between h' and h will be neglected.

49. Saturated Steam.—When steam is in contact with water as it is in a steam boiler it is found that for any particular pressure there is a definite corresponding temperature and also a definite corresponding density. Such steam is called *saturated steam.* For example, if the pressure is 78 lb. per square inch absolute the temperature is 154·6°C. and a cubic foot of it weighs 0·1786 lb. Now if this steam has its temperature lowered by 1° C. its pressure will fall to 76 lb. per square inch, a portion of it will condense to water, and its density will be reduced to 0·1742 lb. per cubic foot. This would have happened just the same if the saturated steam had been first led away into a closed vessel so as to be no longer in contact with the water and then cooled

1°C. Such a result would not be obtained by lowering the temperature of hot air. A cubic foot of hot air contained by a closed vessel would have its pressure lowered by a reduction of its temperature but its density would remain the same. For saturated steam any one of its various properties, pressure, temperature, density, etc., fixes all the others. Saturated steam at a given pressure has the greatest possible density for that pressure.

The principal properties of saturated steam are given in the Appendix, pp. 572 to 575. These properties are connected by formulæ based on the principles of thermodynamics and the results of experiments. The modern formulæ are somewhat complicated and as they are only required for the construction of steam tables they will not be discussed in this work. Associated with the various elaborate steam tables which have been published in recent years discussions on the formulæ used in their construction will be found. Simple approximate formulæ connecting the properties of steam are of little value. The properties required should be taken from the tables.

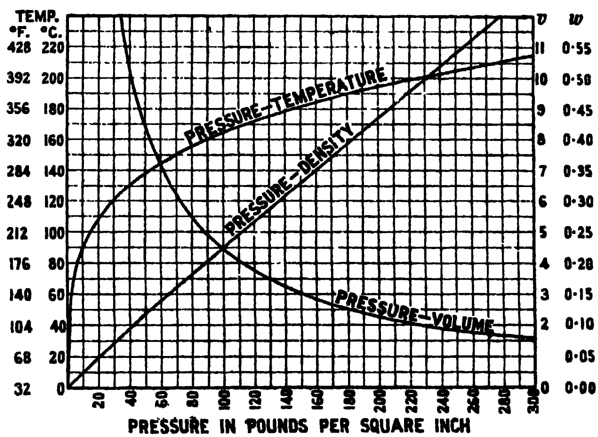


FIG. 42.—Properties of saturated steam.

An inspection of the diagrams, Figs. 42 and 43, will give a general impression of the relations between the properties of saturated steam. In Fig. 42 the abscissæ represent the pressure (p), while the ordinates represent the temperature (t), the specific volume (v), and the density (w), to the scales given. v is in cubic feet per pound, and w is in pounds per cubic foot. w is of course the reciprocal of v.

The form of the pressure-volume curve suggests that the pressure and volume may be related approximately by the formula $pv^n = C$, where C is a constant, and it would form a very good exercise for the student to apply the method of Art. 35, p. 31, to this case. It will be seen that the relation between the pressure and density is roughly a linear one. As to the pressure-temperature curve it will be noticed that the temperature increases much more rapidly with increase of pressure at low than at high pressures.

Coming now to Fig. 43, the abscissæ represent the temperature (t),

while the ordinates represent the sensible heat (h), the latent heat (L), and the total heat (H), all to the same scale. Beyond the temperature 300° C. (572° F.), corresponding to a pressure of about 1250 lb. per square inch, the forms of the curves are uncertain but are probably as shown dotted. At the critical temperature t_c the latent heat is nothing

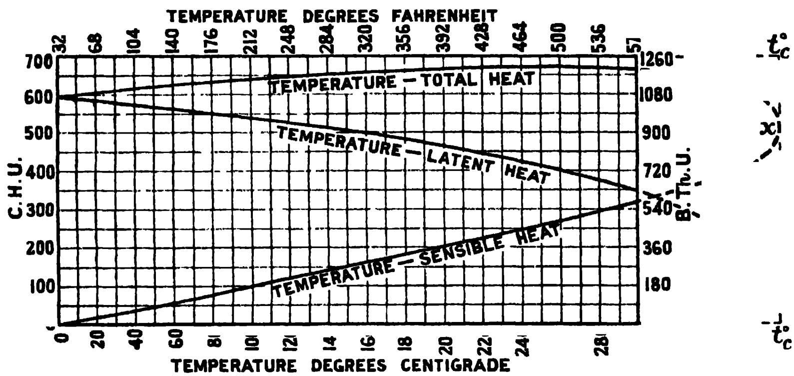


FIG. 43.—Properties of saturated steam.

and the sensible and total heat curves join up at a point x. The critical temperature of steam has generally been taken as 365° C. (689° F.). Professor Goodenough[1] has recently estimated it to be 374·6° C. (706·3° F.).

It will be observed that the three heat curves are fairly flat for the greater portion of their length and that the difference between the maximum total heat and the total heat at the freezing temperature is only about 75 C.H.U. (135 B.Th.U.).

50. Wet Steam—Dryness Fraction.—The steam in the steam space of a boiler generally contains water mixed with it, the water being in a finely divided state forming a mist which is mixed with the steam. Steam in this condition is called *wet steam.* The conditions which favour the production of wet steam are a small steam space in the boiler and rapid evaporation per square foot of area of free water surface. The production of wet steam is called *priming* but the term priming is frequently reserved for cases where the steam produced is very wet. After leaving the boiler saturated steam may become wet and wet steam may become more wet through the loss of heat.

If 1 lb. of wet steam contains x lb. of pure saturated steam, the remainder $1 - x$ being water, then x is called the *dryness fraction* of the wet steam. The *quality* or *dryness* of steam is given by its dryness fraction.

Heat added to wet steam causes water in it to evaporate and when all the water is evaporated but the temperature is not raised the wet steam is said to have been dried. Pure saturated steam is also called *dry saturated steam.*

Considering 1 lb. of wet steam, its total heat is evidently the heat of the water part plus the heat of the steam part and is written

[1] See Appendix, p. 575.

$(1 - x)\,h + x\,(L + h) = h + xL$. The volume is xv, neglecting the volume of the water which is $(1 - x)\,u'$ and is exceedingly small.

51. Superheated Steam.—If the steam after it leaves the steam space in the boiler be passed through tubes which are heated by furnace gases the first effect on the steam is to dry it if it is not already dry. The temperature of the steam then rises and *superheated steam* is the result. Under the usual conditions of superheating the pressure of the steam is not altered because there is free communication between the steam space in the boiler and the interior of the superheater. The superheating therefore takes place under constant pressure.

The greater the amount of superheating the more nearly does the steam acquire the properties of a perfect gas. The amount of superheating is measured by the rise in temperature of the steam above its temperature of saturation.

The advantages which superheated steam possesses over saturated steam will be considered later.

52. Specific Heat of Superheated Steam.—The specific heat of superheated steam varies with the pressure and with the degree of superheating, increasing with the pressure and diminishing with the amount of superheating. For a given pressure the specific heat is greatest at the saturation temperature and it diminishes most rapidly in the neighbourhood of that temperature as the superheating increases.

The best authorities are not yet in agreement as to the values of the specific heat of steam, the disagreement being greatest at saturation temperatures. A general impression of the variation of the specific heat with pressure and amount of superheat will be obtained from an inspection of Fig. 44 which exhibits approximately the specific heat of superheated steam for various pressures and various degrees of superheat.

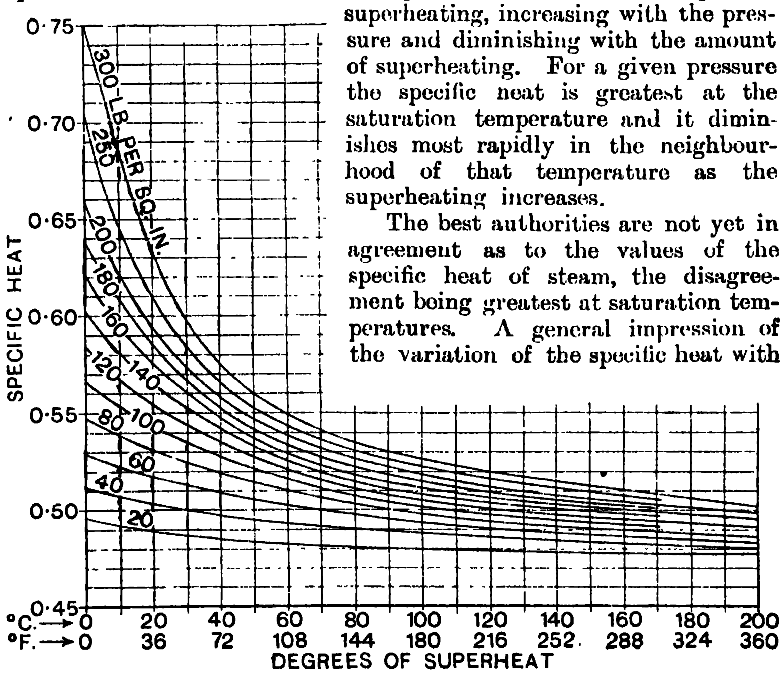


Fig. 44.—Specific heat of superheated steam.

53. Total Heat of Superheated Steam.—The heat required to raise the temperature of 1 lb. of steam from its saturation temperature

t_s to a temperature t, at constant pressure, is $k_p (t - t_s)$, where k_p is the *mean* specific heat at constant pressure between the temperatures t_s and t. The total heat of the steam superheated to the temperature t is therefore $H + k_p (t - t_s)$, where H is the total heat of the dry saturated steam. Tables giving the total heat of superheated steam at various pressures for various degrees of superheat will be found in the Appendix, p. 576.

The total heat of superheated steam, at various pressures, is plotted against degrees of superheat in Fig. 45. This diagram shows clearly that, for a given pressure, as the temperature is increased the amount of heat added per degree increase of temperature becomes more nearly

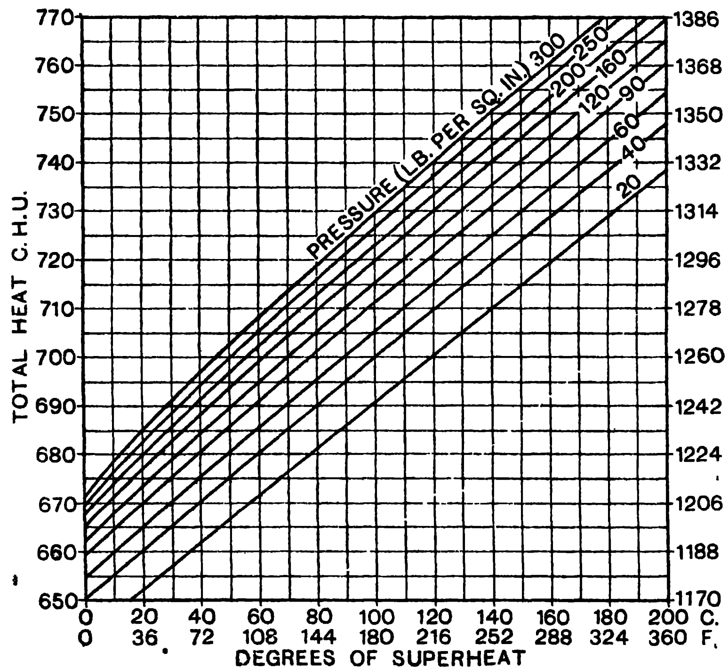


FIG. 45.—Total heat of superheated steam.

constant. Also, the heat added per degree of superheat is not greatly affected by pressure except at low degrees of superheat and then only for high pressures.

54. Specific Volume of Superheated Steam.—Values of the specific volume of superheated steam at various pressures and for various degrees of superheat are given in the Appendix, p. 577. The variations of specific volume with degrees of superheat for various pressures are also exhibited in Fig. 46.

It will be seen that, for a given pressure, the relation between the specific volume and degrees of superheat is approximately a linear one. For some problems it may be sufficiently accurate to assume that, at a given pressure, the volume of superheated steam is proportional to its

absolute temperature, which of course is assuming that it behaves like a perfect gas. Expressed in symbols this means that $V_1 : V_2 :: T_1 : T_2$, where V_1 and V_2 are the volumes of a given weight of superheated steam at absolute temperatures T_1 and T_2 respectively.

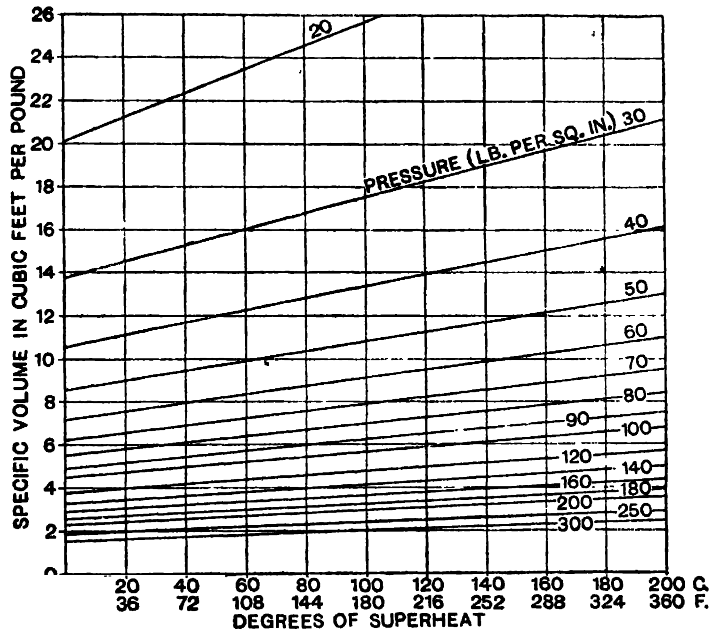


FIG. 46.—Specific volume of superheated steam.

55. Entropy.—The conception of *entropy* is one which is extremely useful in the study of heat engines. The difficulty which the student experiences when this term is first encountered is probably due to the fact that entropy is not a physical property of matter which may be measured directly like weight, volume, temperature, etc.

Probably the simplest way to grasp the conception of entropy is to consider a diagram in which ordinates represent the variation of the temperature of a substance to which heat is being added, or from which heat is being subtracted, and the area represents the heat added, or subtracted; then the abscissæ represent entropy. Such a diagram is called a *temperature-entropy diagram.*

The simplest form of the temperature-entropy diagram is a rectangle such as ABCD, Fig. 47, in which AB represents the absolute temperature T of a substance of unit weight to which q units of heat are added at constant temperature, as for instance when water is boiling. The base AD represents the increase in the entropy of the substance and is denoted by ϕ. The area of the diagram is AB $\times$ AD $= T\phi = q$. In stating that the area of the diagram is equal to q heat units, the unit of area may be taken as that of a rectangle whose height on the temperature scale is 1° and whose base on the entropy

scale is 1 unit of entropy, or it may be a rectangle whose height is 10° and whose base is 1–10th of a unit of entropy, or a rectangle of any other dimensions such that its height measured on the temperature scale multiplied by its base measured on the entropy scale is equal to unity.

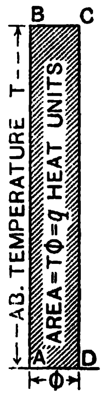


FIG. 47.

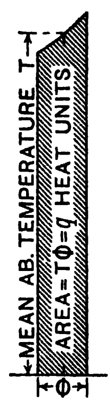


FIG. 48.

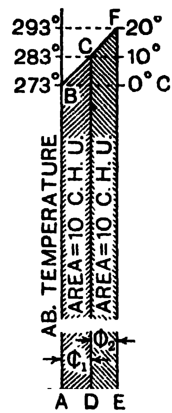


FIG. 49.

If during the operation of adding heat to the substance the temperature rises as shown in Fig. 48 then the product $T\phi$ is still equal to q, the heat added, if T is the mean absolute temperature during the process.

A numerical example showing how the increase in entropy may be calculated when a substance is heated will now be taken. A pound of water at 0° C. or 273° C. absolute has its temperature raised 10° C. Assuming that the specific heat of the water is constant and equal to unity the amount of heat added is 10 C.H.U. and the temperature entropy diagram ABCD, Fig. 49, will have an area representing 10 C.H.U. of heat. Assuming next that the mean absolute temperature or mean height of the diagram is $\frac{273 + 283}{2} = 278$, then if ϕ_1 is the increase in entropy due to the addition of 10 units of heat, $278\phi_1 = 10$. Therefore $\phi_1 = \frac{10}{278} = 0{\cdot}0360$. Now let the temperature be raised another 10° by the addition of another 10 units of heat, and let ϕ_2 denote the further increase in entropy. Taking the mean absolute temperature during the second operation to be $\frac{283 + 293}{2}$ $= 288$, then $288\phi_2 = 10$, and $\phi_2 = 0{\cdot}0347$. The increase in entropy for the two steps is $\phi_1 + \phi_2 = 0{\cdot}0707$.

In the above calculations two assumptions have been made which are not strictly justified. The first assumption was that the specific heat of water is unity, which is not quite true, but this may be corrected by multiplying the rise in temperature by the mean specific heat for that rise to obtain the true amount of heat added. The second assumption was that the mean temperature during the addition of the heat was the arithmetical mean of the temperatures at the beginning and end of the step, which is equivalent to assuming that BC and CF are straight lines. That this is not strictly true is evident when it is observed that for equal increments of temperature the strips such as ABCD and CDEF have equal areas but different mean heights, hence they must have different widths. BC and CF cannot therefore have the same slope. Hence, if in determining the increase in entropy between

0° C. and 20° C. four steps had been taken instead of two the top boundary line would have consisted of four lines of different slopes instead of two. It therefore follows that taking indefinitely small steps the top boundary line would become a curve of increasing slope as the temperature is raised.

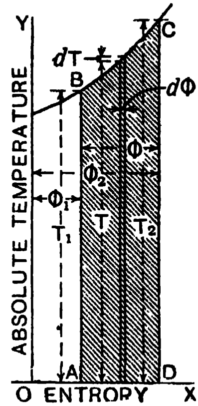


Fig. 50.

An exact mathematical expression for the increase in entropy for a considerable range of temperature is found as follows. Referring to Fig. 50, let the absolute temperature of 1 lb. of the substance be raised from T_1 to T_2, there being no change in the state of the substance between these temperatures. ABCD is the temperature-entropy diagram for the process. Let k be the specific heat of the substance and let this specific heat be constant for the range of temperature T_1 to T_2. Take an indefinitely narrow vertical strip of the diagram where the temperature is T and let the width of this strip be $d\phi$. This quantity $d\phi$ is the indefinitely small increase which takes place in the entropy while the temperature increases by the indefinitely small amount dT.

The heat added for the increase of temperature dT is kdT and

$\mathrm{T}d\phi = kd\mathrm{T}$. Therefore $d\phi = \dfrac{kd\mathrm{T}}{\mathrm{T}}$

Let ϕ be the whole change in entropy between the temperatures T_1 and T_2, then

$$\phi = \int_{T_1}^{T_2} d\phi = \int_{T_1}^{T_2} \frac{kd\mathrm{T}}{\mathrm{T}} = k(\log_e T_2 - \log_e T_1)$$

If O is selected as the point of zero entropy then the entropy at A is $\phi_1 = k\ (\log_e T_1 - \log_e T_0)$ and the entropy at D is

$$\phi_2 = k\ (\log_e T_2 - \log_e T_0)$$

where T_0 is the absolute temperature at O.

The point from which entropy is measured is 0° C. (32° F.), the same as that from which the total heat of water is measured, but just as water at 0° C. does contain heat so it has entropy although the total heat and entropy of water at 0° C. are taken as 0·0.

56. Temperature-Entropy Chart for Water and Steam.—The process of heating a pound of water from the freezing temperature to its boiling point and then evaporating it and finally superheating the steam produced is well shown by a temperature-entropy diagram. Starting at the point A, Fig. 51, as the heat is applied to the water its temperatare rises and its entropy increases, and up to the boiling temperature t the process is represented by the curved line AB. The ordinate of any point in AB represents the temperature of the water and the abscissa represents its entropy above that of the water at the freezing temperature, and for any selected temperature the corresponding entropy may be calculated as explained in the preceding Art. or it may be taken directly from the steam table the Appendix.

Between the freezing and the boiling points the absolute temperature rises from OA to MB, the entropy added is OM, and the heat given to the water is the area of the figure OABM. During the boiling process the temperature remains constant but the entropy increases and the change during the evaporation of the water is represented by the horizontal line BC. The area of the rectangle MBCN represents the heat added during evaporation, which is of course the latent heat L. The increase in entropy is MN and since BM × MN = L it follows that $MN = \frac{L}{BM}$. For example, if the boiling temperature t is 175° C.

$$BM = 273 + 175 = 448,$$

and L is 486·3 C.H.U.

$$\text{Hence MN} = \frac{486\cdot3}{448} = 1\cdot085.$$

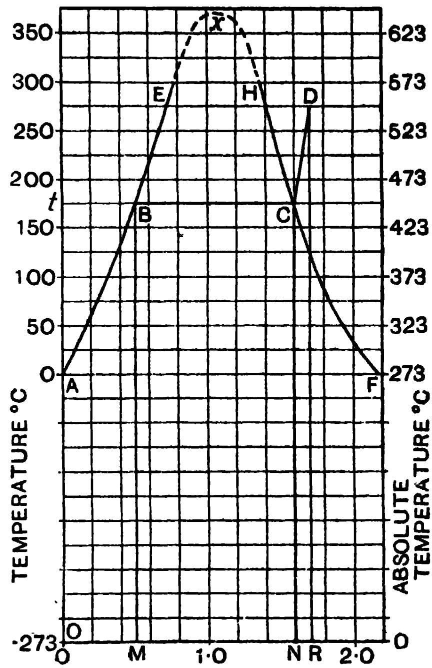


FIG. 51.

Continuing the application of heat, the steam which has been produced will now become superheated under constant pressure, the temperature will rise, and the entropy will increase, and this superheating process will be represented by the curved line CD. The area of the figure CNRD under CD will represent the heat added in superheating the steam and NR will represent the increase in entropy. For example, for 100° C. of superheating the total heat of the superheated steam is 716·8 C.H.U. (by interpolation from the table in the Appendix, p. 576). The total heat of saturated steam at 175° C. is 663·2 C.H.U. The heat added during the superheating process is therefore 716·8 − 663·2 = 53·6 C.H.U. The mean specific heat of the steam for 100° C. of superheat is $\frac{53\cdot6}{100} = 0\cdot536$. CN = 448 and DR = 548. Hence

$$NR = 0\cdot536(\log_e 548 - \log_e 448) = 0\cdot108.$$

The entropy of superheated steam may also be taken directly (or by interpolation) from the table in the Appendix, p. 577.

The curve ABE which shows the relation between the temperature of the water and its entropy is called the *water line*, and the curve FCH, any number of points in which may be found as described for the point C, is called the ***saturated-steam line*** or the ***saturation line***, and

these two curves together with superheated steam curves such as CD and certain other curves to be discussed presently form a *temperature-entropy chart* for water and steam.

If the water line and the saturation line be carried high enough they join and form a continuous curve at x, the critical point at the critical temperature of steam. The exact forms of the water and saturation lines above 300° C. (572° F.) are at present uncertain but this part of the chart is probably as shown by the dotted curve.

The part of the temperature-entropy chart for water and steam which is required in practice in the study of the steam-engine lies between the temperatures 35° C. (95° F.) and 220° C. (428° F.), but in the superheated steam region the chart should extend to 400° C. (752° F.).

Quality or *constant dryness lines* may now be considered. Referring to Fig. 52, ABE is the water line and FCH is the saturation line of a temperature-entropy chart. As already explained, a horizontal line BC terminated by the water and saturation lines represents the process of evaporating a pound of water at a definite temperature, and the point B may be said to represent a pound of water at that temperature while the point C represents a pound of dry saturated steam at the same temperature. It therefore follows that an intermediate point in BC will represent a pound of a mixture of water and saturated steam, that is, a pound of wet steam.

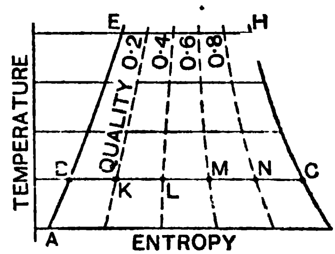


FIG. 52.

In moving from B towards C the distance travelled represents the increase in entropy, and since the temperature is constant the heat added is proportional to the increase of entropy. But after evaporation begins the amount of water evaporated is proportional to the heat added. Hence if BC be divided into, say, five equal parts at the points K, L, M, and N these points will represent 1/5, 2/5, 3/5, and 4/5 of a pound of saturated steam, and 4/5, 3/5, 2/5, and 1/5 of a pound of water respectively. Stated in another way, the points K, L, M, and N represent a pound of wet steam at a definite temperature and of dryness fractions or qualities 0·2, 0·4, 0·6, and 0·8 respectively. Dividing other horizontal intercepts between the water and saturation lines each into the same number of equal parts, and joining corresponding points by fair curves as shown, lines of quality 0·2, 0·4, 0·6, and 0·8 are obtained.

Any point in a line of quality x represents a pound of wet steam of that quality at a temperature given by the level of the point on the temperature scale.

Constant volume lines have now to be considered. Referring to Fig. 53, ABE and FCH are portions of the water and saturation lines of a temperature-entropy chart. BC is the horizontal intercept between the water and saturation lines at the temperature 160° C. Since distances from B along BC are proportional to weight of water evaporated, these distances will also be proportional to the volume of

steam produced, the temperature and therefore the pressure being constant. The distance BC may be taken as representing the volume of a pound of dry saturated steam at the temperature 160° C., namely, 4·92 cubic feet. If N is a point in BC such that BN represents 4 cubic feet of steam then $BN = \frac{4}{4 \cdot 92} \times BC$. Measuring BC with any scale, BN can be calculated and then measured off with the same scale. Dividing BN into four equal parts, points representing 1, 2, and 3 cubic feet are found. Applying the above method to other horizontal intercepts between the water and saturation lines at definite temperatures, other points representing 1, 2, 3, 4, etc., cubic feet are found, and joining corresponding points by fair curves as shown, lines of constant volume are determined.

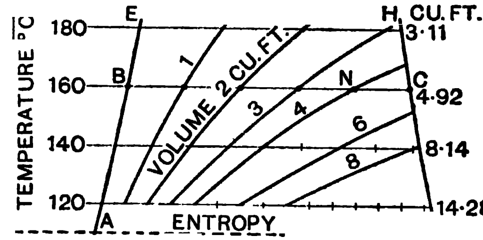


Fig. 53.

Any point in a line of constant volume v represents a pound of wet steam having a volume v cubic feet at a temperature given by the level of the point on the temperature scale.

In what has been said above about the volume of wet steam the volume of the water part has been neglected and this is justified because of the very small volume of the water compared with that of the steam.

A fairly complete temperature-entropy chart for water and steam is given in Fig. 54 but a chart suitable for use in practice should be much larger and the quality lines and the lines of constant volume should be more numerous; the horizontal temperature lines and the vertical entropy lines should also be more numerous.

An excellent exercise for the student is to construct this chart and the other charts which follow in this chapter on large sheets of squared paper of the best quality procurable. Paper squared in centimetres and millimetres is the most suitable.

In using a chart the problem to be solved should be worked on a sheet of tracing paper fixed over the chart.

A temperature-entropy chart for superheated steam is given in Fig. 55. The remarks made in connection with Fig. 54 also apply to Fig. 55. Points in the constant pressure lines and points in the lines of constant total heat may be plotted directly or by interpolation from steam tables.

57. Total Heat-Entropy Chart.—Another form of chart which for some purposes is more useful than the temperature-entropy chart is that in which the co-ordinates are total heat and entropy instead of temperature and entropy. A portion of a *total heat-entropy chart* for steam, also known as the *Mollier chart,* is shown in Fig. 56 to a small scale. Such a chart for general use may have a range in entropy from 1·3 to 2·1 and a range in total heat from 420 C.H.U. (756 B.Th.U.) to 820 C.H.U. (1476 B.Th.U.).

In constructing this chart points in the saturation line are first plotted. Lines of constant pressure are then determined. In the

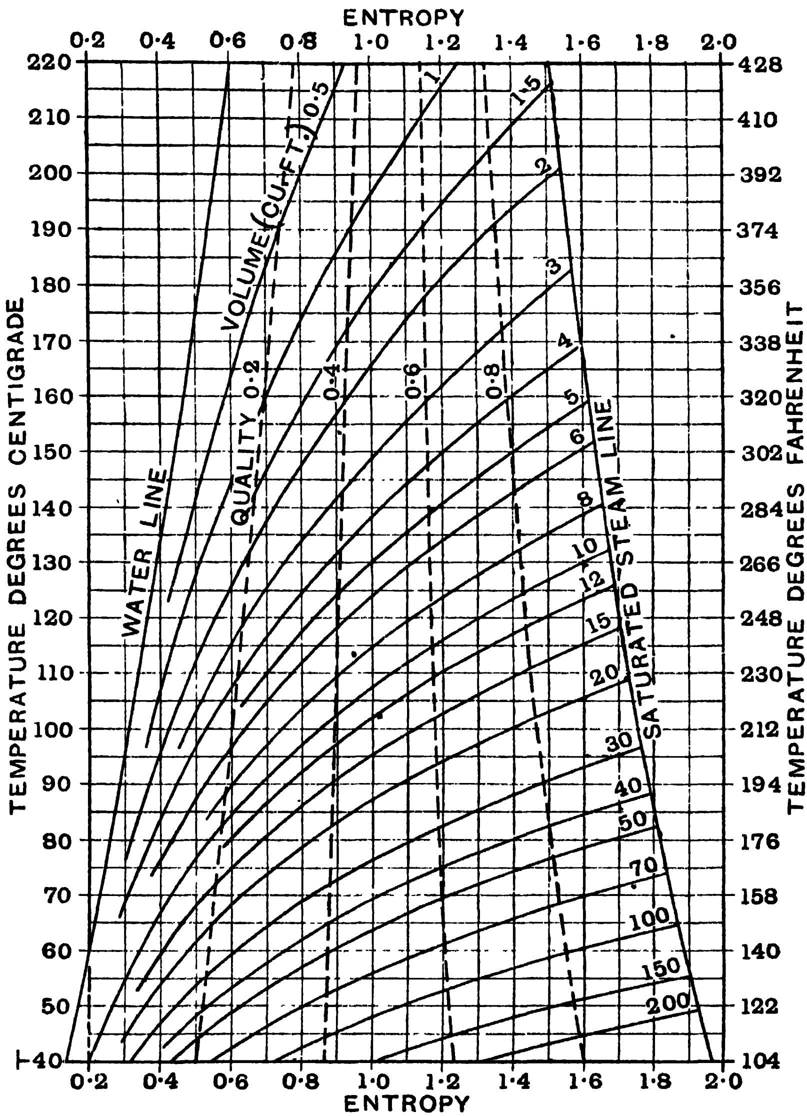


FIG. 54.—Temperature-entropy chart for water and steam.

wet steam region below the saturation line the constant pressure lines are straight lines. This follows from the consideration that for wet steam constant pressure is associated with constant temperature and a reference to the temperature-entropy chart makes it clear that

at constant temperature equal increments of heat are accompanied by equal increments of entropy. Hence the relation between total heat and entropy for wet steam at constant pressure is a linear one.

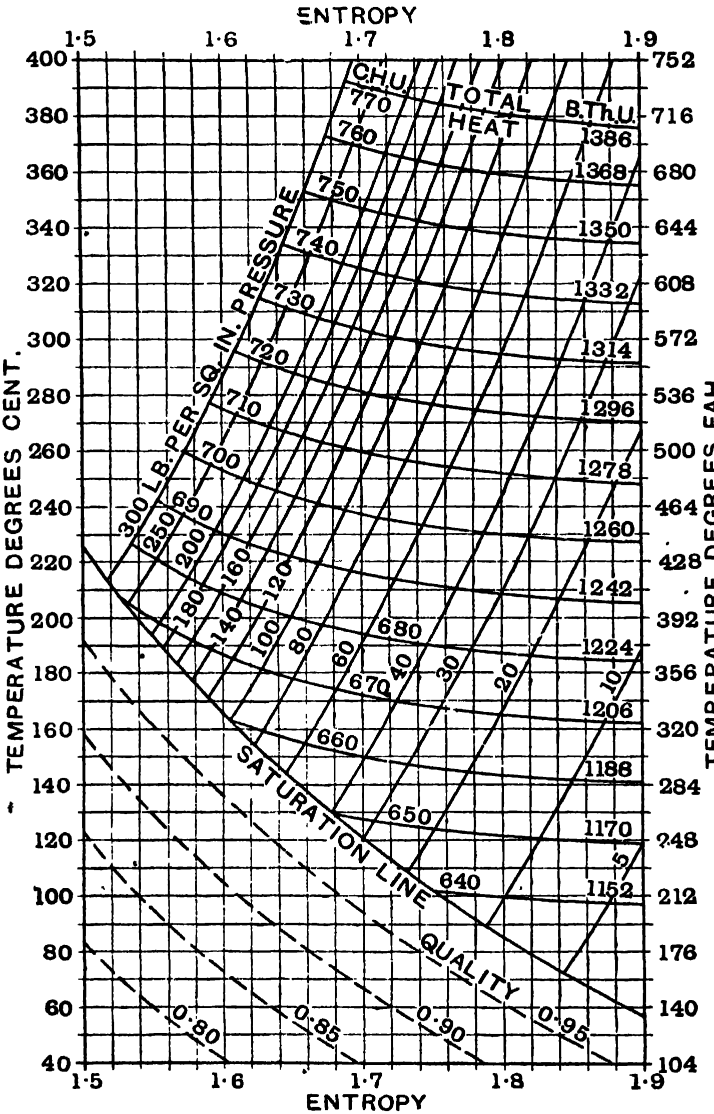


FIG 55.—Temperature-entropy chart for superheated steam.

In the superheated steam region points in the constant pressure lines are plotted from the total heat and entropy tables for superheated steam. By selecting the points for the pressure lines in

groups at constant degrees of superheat the superheat lines are determined at the same time as the pressure lines. Instead of lines of constant superheat lines of constant temperature may be used

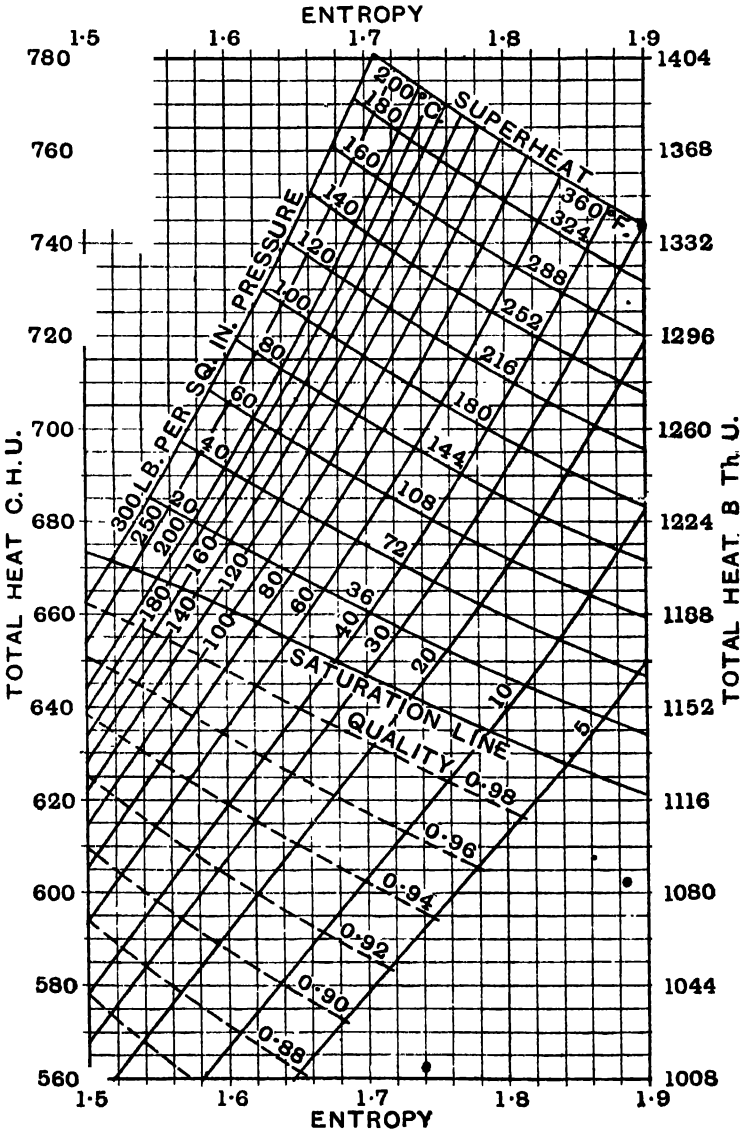


FIG. 56.—Total heat-entropy chart for steam.

In regard to the quality lines it should be noticed that, for equal differences in quality, the quality lines cut any selected pressure line at equidistant points.

An actual chart for general use should be much larger than the one shown in Fig. 56, and the lines of constant pressure, the lines of constant superheat, and the lines of quality should be more numerous.

58. Total Heat-Pressure Chart.—Still another form of chart in use is that in which the co-ordinates are total heat and pressure. This chart has on it lines of constant volume, lines of constant quality, and lines of constant superheat. Lines of constant temperature may be substituted for lines of constant superheat.

Lines of constant volume and lines of constant temperature are straight lines which change in direction abruptly where they cross the saturation line. Lines of constant superheat and lines of quality are curved.

Sometimes this chart is drawn with the pressure scale varying so that the corresponding saturation temperatures are to a uniform scale. The advantage of this modification is that sufficient constant volume lines at low pressures may be inserted without overcrowding. The disadvantage is that the lines mentioned above as being straight are now slightly curved, except the constant temperature lines in the wet steam region which remain perpendicular to the pressure base.

59. Adiabatic Expansion of Steam.—It has already been stated (Art. 32, p. 30) that in adiabatic expansion no heat as heat is given to or taken from the expanding gas. This does not necessarily mean that the amount of heat in the gas remains constant during the expansion. If work is done during adiabatic expansion this work is done at the expense of heat in the gas, but no heat leaves the gas to appear immediately as heat in another substance.

Three cases of adiabatic expansion of steam have to be considered. In the first case the steam expands against a resistance, as in the cylinder of a reciprocating steam engine, and work is done at the expense of heat in the steam, the amount of heat disappearing being equivalent to the work done. Since there is no transference of heat as heat during the expansion there is no change in the entropy; this is evident from the definition of entropy given in Art. 55, p. 59.

When there is no change in the entropy of a gas during expansion the expansion is said to be *isentropic.* Isentropic expansion is also adiabatic, but adiabatic expansion is not necessarily isentropic. Isentropic expansion of steam is considered in the next Article.

In the second case the work done by the steam as it expands adiabatically appears as kinetic energy giving increased velocity to the steam, as when it issues through a nozzle, and again there is no change in the entropy of the steam and the expansion is therefore isentropic in this case as in the first. The calculation of the kinetic energy and velocity acquired is considered in Arts. 288 and 289, p. 398 and p. 402.

In the third case the net result is that no work is done and there is no change in the total heat in the steam, as when steam is throttled or wire-drawn. In this case the entropy of the steam is increased and the expansion is therefore not isentropic. This case is considered in Art. 62, p. 73.

In general, when adiabatic expansion is mentioned without qualification isentropic expansion is understood.

60. Isentropic Adiabatic Expansion of Steam.—By means of the temperature-entropy chart for water and steam some points in the behaviour of steam are made very clear and a number of problems may be readily solved by the use of this and other steam charts. A study of the adiabatic expansion of steam with the aid of the temperature-entropy chart will be of interest in this connection.

It is evident that the isentropic adiabatic expansion of steam must be represented on the temperature-entropy chart by a straight line perpendicular to the entropy base. Fig. 57 shows the portion of a temperature-entropy chart for water and steam lying between the constant temperature lines 180° C. and 100° C., ABE being the water

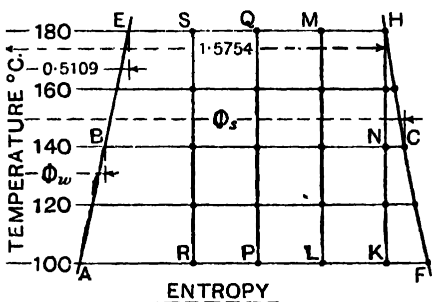


FIG. 57.

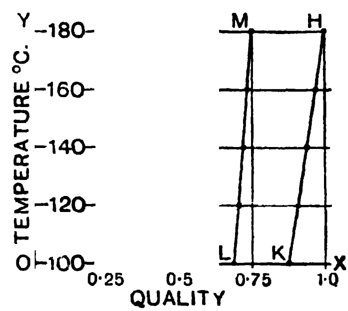


FIG. 58.

line and FCH the saturation line. The vertical straight lines HK, ML, QP, and SR are adiabatics representing the expansion of 1 lb. of steam of various degrees of initial dryness. The points S, Q, and M divide EH into four equal parts and the initial dryness of the steam is therefore 1·0 for HK, 0·75 for ML, 0·5 for QP, and 0·25 for SR.

In studying these adiabatics it will be convenient first to tabulate certain data from the steam tables, namely, the temperature t in degrees Centigrade, the pressure p in lb. per square inch, the specific volume v in cubic feet, the entropy ϕ_w of the water, and ϕ_s the entropy of the dry saturated steam. Certain results to be computed will be added in columns provided for the purpose.

	p	v	ϕ_w	ϕ_s	x_1	x_2	x_3	
180	145·5	3·11	0·5109	1·5754	1·000	0·750	0·500	0·250
160	89·8	4·92	0·4639	1·6136	0·967	0·735	0·504	0·272
140	52·4	8·14	0·4153	1·6562	0·935	0·720	0·506	0·291
120	28·8	14·28	0·3646	1·7038	0·904	0·705	0·507	0·308
100	14·7	26·79	0·3118	1·7568	0·874	0·690	0·506	0·322

The first point to notice is that the quality or dryness of the steam varies during the adiabatic expansion. The symbols x_1, x_2, x_3, and x_4 will denote the quality of the steam for the adiabatics HK, ML, QP, and SR respectively, and these are to be computed and tabulated for the temperatures 160°, 140°, 120°, and 100°; their values for the

temperature 180° are the values at the beginning of the expansion and these have been given.

Consider the adiabatic HK. At the temperature 140° the dryness fraction x_1 is $\frac{\text{BN}}{\text{BC}} = \frac{1{\cdot}5754 - \phi_w}{\phi_s - \phi_w} = \frac{1{\cdot}5754 - 0{\cdot}4153}{1{\cdot}6562 - 0{\cdot}4153} = 0{\cdot}935$. In like manner the values of x_1 at 160°, 120°, and 100° are determined. It is seen that for the adiabatic HK the steam gets wetter, that is, more of it condenses, as it expands.

For the adiabatic ML the constant entropy is—

$$0{\cdot}5109 + 0{\cdot}75(1{\cdot}5754 - 0{\cdot}5109) = 1{\cdot}3093.$$

At 140°, $x_2 = \frac{1{\cdot}3093 - \phi_w}{\phi_s - \phi_w} = \frac{1{\cdot}3093 - 0{\cdot}4153}{1{\cdot}6562 - 0{\cdot}4153} = 0{\cdot}720.$

In like manner the other values of x_2 and also the values of x_3 and x_4 required, are found.

The variation in the quality of the steam as it expands adiabatically is better shown by plotting the results as shown in Fig. 58. It is now more clearly seen that the dryer the steam is at the beginning of the expansion the greater is the amount of condensation which takes place. When the initial dryness is 0·5 it is seen that no condensation takes place during expansion but instead the steam becomes slightly dryer. With wetter steam the drying effect of adiabatic expansion is seen to be more pronounced.

The variation in the volume of the expanding steam may now be considered. The symbols v_1, v_2, v_3, and v_4 will denote the volumes in cubic feet of 1 lb. of wet steam for the adiabatics HK, ML, QP, and SR respectively, and these are to be computed and tabulated for the temperatures 180°, 160°, 140°, 120°, and 100°.

At any point the volume of the wet steam is equal to the dryness fraction at that point multiplied by the specific volume of the dry saturated steam at the same temperature. For example, at the point N in the adiabatic HK, Fig. 57, where the temperature is 140° it has been found that the dryness fraction is 0·935, and for the temperature 140° $v = 8{\cdot}14$, therefore for the point N, $v_1 = 0{\cdot}935 \times 8{\cdot}14 = 7{\cdot}61$.

The required values of v_1, v_2, v_3, and v_4 are here tabulated.

t	v_1	v_2	v_3	v_4
180	3·11	2·33	1·56	0·78
160	4·76	3·62	2·48	1·34
140	7·61	5·86	4·12	2·37
120	12·91	10·07	7·24	4·40
100	23·41	18·49	13·56	8·63

It is now possible to draw the "pv" diagrams for the various adiabatics which have been considered. The values of v_1, v_2, v_3, and v_4 have been plotted against the corresponding pressures in Fig. 59 and the fair curves drawn through the points obtained are the pv curves for the various adiabatics. For comparison the pv curve for the saturated steam line has been added.

It is evident that in Fig. 59 if a horizontal line BNC be drawn cutting the line of zero volume at B, the saturation line at C, and an adiabatic at N, the dryness fraction of the steam represented by the point N is BN/BC.

For the pv curve of an adiabatic for wet steam pv^n = a constant is approximately true and the value of n may be found as explained in Art. 35, p. 31, by plotting $\log p$ and $\log v$; this has been done in Fig. 60. The values of n are approximately as follows:—For HK, $n = 1{\cdot}136$; for ML, $n = 1{\cdot}107$; for QP, $n = 1{\cdot}060$; and for SR,

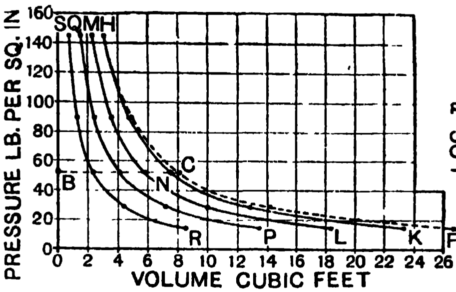


FIG. 59.

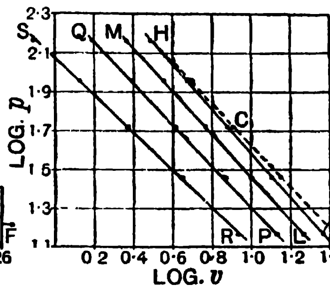


FIG. 60.

$n = 0{\cdot}955$. In addition the saturation line HCF has been dealt with in the same way and n found to be $1{\cdot}068$.

The student will find it to be a very profitable exercise to verify all the foregoing tabulated results and construct carefully the diagrams in Figs. 57 to 60, drawing them say four times the size shown.

Another example which the student should work out in full is here suggested. In Fig. 61 a part of the temperature-entropy chart is shown, FCH being the saturation line. CD is the constant pressure line for superheated steam, the pressure being 160 lb. per square inch. The amount of superheat at D is 80° C. and the temperature at D is therefore 264·2° C. The vertical straight line DN represents the adiabatic expansion of the superheated steam down to 100° C. (14·7 lb. per square inch). KL is the constant pressure line for superheated steam of 100 lb. per square inch pressure. The adiabatic DN cuts KL at R and the saturation line at S. The volume of steam at D, taken from the table on p. 577, is 3·50 cubic feet, and the entropy from the table on p. 577 is 1·6572. It is required to find the volumes at R, S, and N, also the pressure at S. The logs of the pressures and volumes at D, R, S, and N are to be plotted as in Fig. 60. It will be found that the index n in the formula pv^n = a constant is about 1·316 for the part DS of the adiabatic and about 1·136 for the part SN.

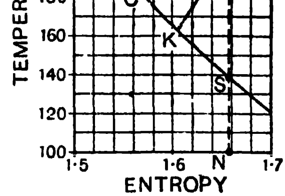


FIG. 61.

The steam remains superheated between D and S but condensation begins at S and continues to the end of the expansion. The temperatures at R and S may be found directly from the student's chart if it is a good one or they may be found by interpolation from the steam tables. The dryness fraction at N is easily calculated and from that

and the specific volume of saturated steam at 100° C. the volume at N is readily found.

61. Work done when Expansion is Isentropic.—It has been shown in the preceding Article how the pv curve may be constructed from the isentropic expansion line on the temperature-entropy chart, also how the index n in the formula $pv^n = $ a constant may be found. It will be instructive to examine now the pv diagram and compute from it the work done per pound of steam used.

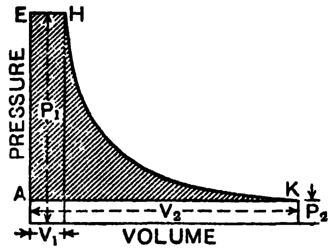


FIG. 62.

Referring to Fig. 62, HK is the adiabatic expansion curve for one pound of steam between the pressures P_1 and P_2 and volumes V_1 and V_2, the co-ordinates being pressure and volume. Assume that the steam is used in the cylinder of a steam engine. When the steam begins to expand its volume is V_1 but before expansion begins this volume V_1 of steam at pressure P_1 has to be introduced into the cylinder and during this operation the work done is P_1V_1. During expansion the work done (Art. 37, p. 35) is $\frac{P_1V_1 - P_2V_2}{n-1}$. During the return stroke the steam exhausts at the constant pressure P_2 and the net work done on one side of the piston during two consecutive strokes is

$$P_1V_1 + \frac{P_1V_1 - P_2V_2}{n-1} - P_2V_2 = \frac{n}{n-1}(P_1V_1 - P_2V_2).$$

Applying the foregoing formula to the case in the preceding Article where the initial pressure is 145·5 lb. per sq. in., final pressure 14·7 lb. per sq. in., initial volume 3·11 cubic feet, and the volume at the end of the expansion 23·41 cubic feet, the expansion line on the temperature-entropy diagram, Fig. 57, being HK, from which n was found to be 1·136 :

$$\text{Work done per lb. of steam} = \frac{n}{n-1}(P_1V_1 - P_2V_2)$$

$$= \frac{1{\cdot}136}{1{\cdot}136 - 1} \times 144\ (145{\cdot}5 \times 3{\cdot}11 - 14{\cdot}7 \times 23{\cdot}41) = 130360 \text{ ft.-lb.}$$

The heat equivalent of this work is $\frac{130360}{1400} = 93{\cdot}1$ C.H.U.

The dryness fraction of the steam at K (Fig. 57) was found to be 0·874. The total heat of the steam at K is therefore

$$0{\cdot}874 \times 539{\cdot}2 + 100 = 571{\cdot}3 \text{ C.H.U.}$$

The total heat of the steam at H is 664·4 C.H.U. Hence the heat converted into work is 664·4 − 571·3 = 93·1 C.H.U. which agrees with the result obtained from the pv diagram.

The student would do well to deal with all the cases in the preceding article in the above manner and tabulate the results. It will be observed that the index n may be calculated from the equation $\frac{n}{n-1}(P_1V_1 - P_2V_2) = QJ$, where Q is the heat converted into work

determined from the temperature-entropy diagram and J is the mechanical equivalent of heat.

Referring now to the temperature-entropy diagram Fig. 63. The chart is there shown down to the absolute zero line of temperature OX. The heat in the steam at H where the expansion begins is represented by the figure OGAEHI, and the heat in the wet steam at K where the expansion ends is represented by the area OGAKI. The hatched area AEHK therefore represents the difference between the heat in the steam at the beginning and at the end of the expansion and, consequently, the heat converted into work. For the numerical example worked out above the area AEHK represents 93·1 C.H.U.

The diagram AEHK, Fig. 57, or Fig. 63. is a temperature-entropy diagram corresponding to the pressure-volume diagram Fig. 62.

The cycle of operations represented by the diagrams AEHK, Figs. 62 and 63, may be described as follows. A pound of water whose condition is represented by the point A is pumped into the boiler where it is heated, its temperature and pressure being raised. This operation is represented by the line AE. The water then evaporates under constant pressure at constant temperature. This operation is represented by EH. The steam then expands, this expansion being represented by the line HK. The last operation is the condensation of the steam at constant temperature and constant pressure and this is shown by the line KA. The various changes shown by Fig. 62 are changes of pressure and volume and area represents work done, while the changes shown by Fig. 63 are changes of temperature and entropy, and area represents heat expended.

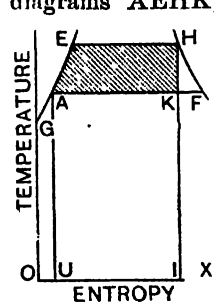


Fig. 63.

62. Wire-Drawing or Throttling of Steam.—A constriction in a passage through which a gas is flowing has the effect of reducing the pressure on the delivery side and therefore of causing an expansion of the gas. Work may be done in the neighbourhood of the constriction in giving eddying or whirling motions to the particles of the gas but this work is reconverted into heat as a steady flow is resumed on the delivery side. The net result is that, if there is no interchange of heat between the gas and the walls of the passage, that is, if the flow is adiabatic, there is no change in the total energy of the gas, and neglecting the kinetic energy of the gas due to its flow the total heat of the gas is unaltered by throttling.

The effect of wire-drawing or throttling on steam is also to alter its quality, which will be best shown by the consideration of three numerical examples, in each of which one pound of steam is throttled from a pressure of 160 lb. per sq. in. to a pressure of 20 lb. per sq. in. The effect of throttling on steam is best studied with the aid of the total heat-entropy chart a portion of which is shown in Fig. 64 where FCHI is the saturation line and AEH and BFK are constant pressure lines for pressures of 160 and 20 lb. per sq. in. respectively.

Example I.—Before throttling, the steam has a dryness fraction of 0·94 and its state is represented by the point A (Fig. 64). The

adiabatic expansion without change of total heat is represented by the horizontal line AB. By interpolation on the chart the dryness fraction of the steam at B, where the expansion ends, is seen to be nearly 0·99.

With the aid of the steam tables the dryness fraction x at B may be calculated as follows:

$$\begin{aligned} \text{Total heat in steam at A} &= 0{\cdot}94\ \mathrm{L} + h \\ &= 0{\cdot}94 \times 478{\cdot}9 + 186{\cdot}5 = 636{\cdot}7\ \text{C.H.U.} \\ \text{Total heat in steam at B} &= x\mathrm{L} + h \\ &= 533{\cdot}6\,x + 109{\cdot}0 = 636{\cdot}7. \\ \text{Therefore}\qquad x &= 0{\cdot}989. \end{aligned}$$

EXAMPLE II.—The steam is superheated 20° C. at the end of the expansion. From the table on p. 576 the total heat is found to be 652·4 C.H.U. The steam at the end of the expansion is therefore represented by the point D (Fig. 64) on the line BFK where the total

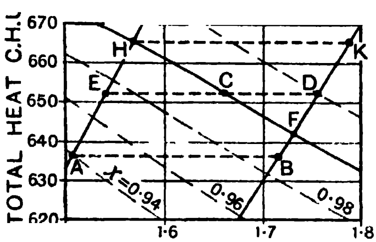


FIG. 64.

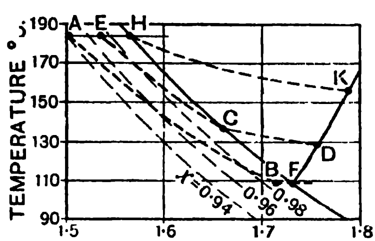


FIG. 65.

heat is 652·4. A horizontal line ECD cutting the initial pressure line AH at E and the saturation line at C represents the expansion in this case. By interpolation on the chart the dryness fraction at E is seen to be rather more than 0·97.

The dryness fraction x at E is found by calculation as follows:

$$x\mathrm{L} + h = 478{\cdot}9x + 186{\cdot}5 = 652{\cdot}4$$

$$\text{Therefore}\qquad x = 0{\cdot}973.$$

EXAMPLE III.—At the beginning of the expansion the steam is dry and saturated. The initial state is represented by the point H (Fig. 64) and the expansion is represented by the horizontal line HK. The total heat at H and at K is 665·4 C.H.U.

By interpolation in the table on p. 576 it is found that the degree of superheat at K is 46·8° C. and the temperature at K is therefore 108·9 + 46·8 = 155·7° C.

The degree of superheat at K may also be found by interpolation between the superheat or temperature curves.

The expansion lines for the foregoing examples are also shown on the temperature-entropy chart in Fig. 65, the same letters being used for corresponding points. The fact that the constant total heat lines are straight on the total heat-entropy chart while they are curved on the temperature-entropy chart makes the former chart the more

convenient one for problems on throttling. It should also be noticed that isentropic expansion lines are straight on both charts.

63. Steam Calorimeters.—The quality or dryness of steam may be determined experimentally by means of a *steam calorimeter*. In practice when it is desired to know the quality of a supply of steam only a comparatively small sample of the whole is available for testing and any want of reliability in determinations of the quality of steam is mainly due to the difficulty of obtaining a true sample.

Most commonly the sample is taken by means of a tube of about half-inch bore projecting into the pipe carrying the steam to be tested, as shown at (*a*) in Fig. 66. This tube AB is closed at its inner end and the sample of steam is taken in through a number of holes drilled in it, these holes being preferably on one side only and facing the

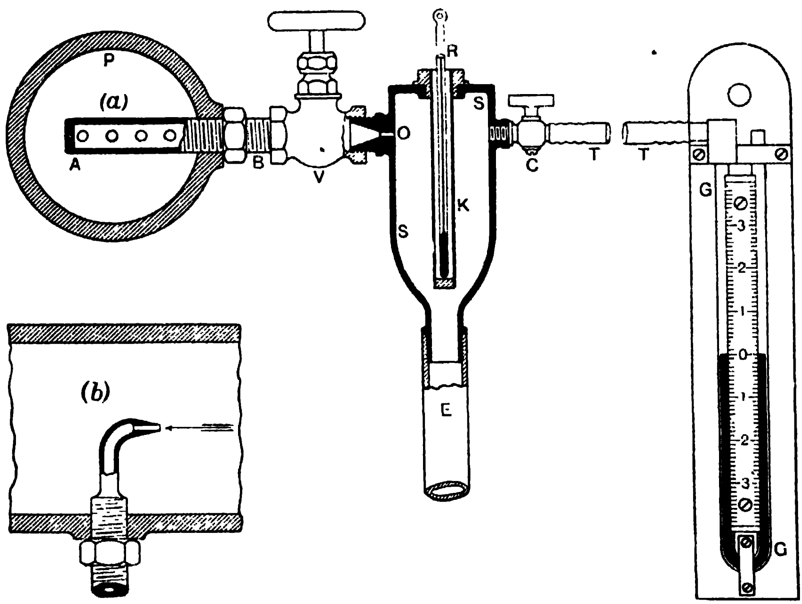


FIG. 66.—Throttling steam calorimeter.

steam current. In the case of a horizontal pipe, water deposited from the steam will be carried along on the bottom surface of the pipe and a truer sample will probably be obtained by inserting the sampling tube from below, taking care that one of the collecting holes is just inside the pipe and facing the steam current.

Professor Unwin has suggested inserting in the centre of the steam pipe a small nozzle facing the current of steam, the nozzle being so proportioned that the velocity of the steam entering into it should be about the same as the general velocity of the steam along the pipe. A design of steam collecting nozzle is shown at (*b*) in Fig. 66.

The *throttling calorimeter* was first introduced by Professor Peabody. Fig. 66 shows Professor Carpenter's modification of the Peabody throttling calorimeter. P is the pipe carrying the steam to be tested.

AB is the sampling tube through which the sample of steam flows to the vessel S which is the principal part of the calorimeter. V is a stop valve. The sample of steam enters the vessel S through a small orifice O and expands to a pressure a little above that of the atmosphere into which it escapes freely through the exhaust pipe E. The pressure of the steam in S is measured by means of the manometer or siphon gauge G which is a glass U-tube containing mercury. G is connected to S by a rubber tube T. Steam to the gauge may be cut off by means of the stop cock C. A deep pocket K, containing cylinder oil, takes the thermometer R which indicates the temperature of the steam in S. The vessel S, the stop valve V, and the exposed part of the sampling tube AB must be carefully lagged to prevent undue loss of heat by radiation.

In using this instrument four observations have to be made. (1) The pressure p_1 or temperature t_1 of the steam in the pipe P. (2) The pressure of the steam in the vessel S as shown by the gauge G in inches of mercury. (3) Reading of thermometer R. (4) The height of the barometer in inches of mercury. Inches of mercury may be converted into pounds per square inch by multiplying by 0·49.

The principle upon which the operation of the throttling calorimeter depends is that the total heat in the steam after throttling is the same as the total heat in the steam before throttling, as shown by the equation

$$h_1 + xL_1 = H_2 + k_p(t_3 - t_2)$$

from which

$$x = \frac{H_2 - h_1 + k_p(t_3 - t_2)}{L_1}$$

where

x = dryness fraction of steam before throttling.
h_1 = sensible heat of steam before throttling
L_1 = latent heat of steam before throttling.
H_2 = total heat of saturated steam at pressure of steam after throttling.
t_2 = temperature of saturated steam at pressure of steam after throttling.
t_3 = temperature of steam after throttling as indicated by thermometer R.
k_p = mean specific heat of steam between temperatures t_2 and t_3 at constant pressure of steam after throttling.

Since the pressure after throttling is in the neighbourhood of that of the atmosphere, and since the specific heat of steam at that pressure is little affected by the temperature, k_p may be taken as 0·48.

The operation of the throttling calorimeter depends on the steam being superheated after throttling and it will fail in its purpose when the steam is so wet before throttling that it is wet after throttling. This means that t_3 must be greater than t_2. Putting $t_3 = t_2$, the value of x which may be determined by the throttling calorimeter must be greater than $\frac{H_2 - h_1}{L_1}$. For example, if the steam has an absolute pressure of 300 lb. per sq. in. its dryness must be greater than 0·93 if it is to be measured with this instrument, and at 100 lb. per sq. in. pressure the dryness must exceed 0·96.

If a throttling calorimeter is supplied with dry saturated steam so that $x = 1$ the instrument may evidently be used to determine the specific heat of superheated steam.

The *separating calorimeter* is quite simple in its action and it may be used for testing steam of practically any degree of wetness. Professor Carpenter's design of separating calorimeter is shown in Fig. 67. The sample of steam to be tested enters the calorimeter at D and passes down through the central passage DF into the perforated metal cup H. The current of steam then reverses and flows as shown by the arrows into the jacket J surrounding the inner chamber C. From the jacket the steam escapes into the atmosphere through the small orifice at the bottom and the exhaust pipe E.

The water in the steam is thrown out in the cup H and collected in the chamber C. A gauge G has two scales on its dial, the inner one showing the pressure of the steam in the jacket while the outer one shows the weight of steam passing through the jacket for each pressure in ten minutes. The graduations on the outer scale are determined by experiment with steam of different pressures, the escaping steam being condensed and weighed, but in using the instrument it is only necessary to take during a ten-minute test the average reading on the outer scale to obtain the weight of steam passing through the jacket.

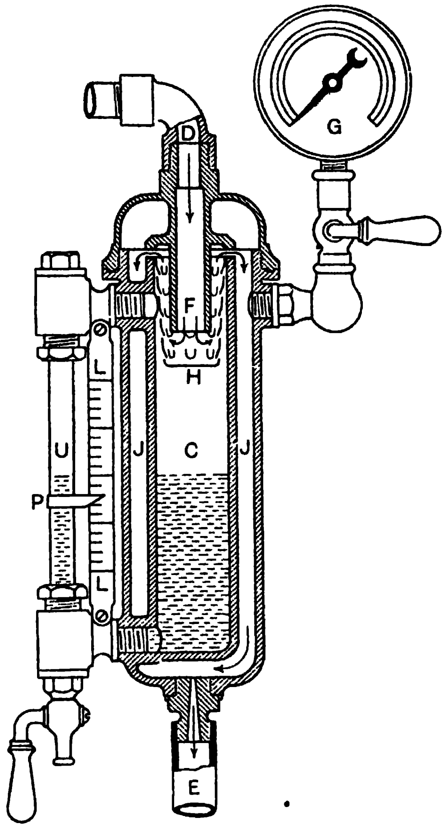


Fig. 67.—Separating calorimeter.

The water separated from the steam is shown by means of the glass gauge U. A pointer P having a frictional grip of the glass tube is set to the water level at the beginning of a test. The change in level of the separated water during a test is read off on the scale L which is graduated in fractions of a pound.

If W is the weight of steam passing through the jacket during a test and w is the weight of water separated in the same time then x, the dryness fraction of the steam tested, is $\frac{W}{W + w}$.

The weight of steam passing through the jacket may also be readily determined by passing the escaping steam into a bucket of water where it is condensed, the increase in the weight of the bucket and its contents giving the value of W. In that case the gauge G is not required, but if used its reading may be taken as a check.

The exposed part of the sampling tube should be properly lagged and if the gauge G is used to read the weight of steam passing through the jacket the instrument itself should also be lagged.

Exercises III

1. How many foot-pounds of work are required to pump 100 lb. of water into a boiler under a pressure of 100 lb. per sq. in.? What is the equivalent of this work in thermal units? Assume that a cubic foot of the water as pumped weighs 62·3 lb.

2. A steam boiler contains 300 cubic feet of water and 300 cubic feet of steam at a pressure of 180 lb. per square inch absolute. Compute the total energy in the water and in the steam in heat units and in foot-tons. Take the weight of a cubic foot of the water at the temperature corresponding to the steam pressure as 54·6 lb.

3. Apply the method of Art. 35, p. 31, to determine the values of n and C in the formula $pv^n = C$ which will make this formula most nearly agree with the given tabulated values of p and v for one lb. of saturated steam.

50	80	110	140	170	200
8·51	5·47	4·06	3·23	2·68	2·30

Having found n and C use the formula to calculate v for the following values of p.—30, 90, 150, and 250.

4. Given that the densities (w) of saturated steam at absolute pressures (p) of 100 and 150 lb. per square inch are 0·2252 and 0·3311 lb. per cubic foot respectively, and assuming that for intermediate pressures the densities are given by the formula $w = a + bp$, find the values of a and b, and then use the formula to calculate the densities for $p = 115$, and $p = 129·5$. Also, from the densities obtained calculate the specific volumes (v) in cubic feet per lb.

5. Determine the amount of heat required to produce 1 lb. of steam, dryness fraction 0·97, absolute pressure 150 lb. per sq. in., from water at 35° C. (95° F.).

6. The heat produced by the combustion of 1 lb. of a certain kind of coal is 7600 C.H.U. (13680 B.Th.U.). This coal is burned in a steam boiler and 70 per cent. of its heat of combustion is utilized in producing steam whose dryness fraction is 0·98 and absolute pressure 180 lb. per sq. in. from water at 50° C. (122° F.). Calculate the weight of steam per lb. of coal used.

7. From a steam engine indicator diagram it was found that when the volume of steam behind the piston was 2·16 cubic feet its absolute pressure was 88 lb. per square inch. From other data it was found that the weight of this steam was 0·54 lb. Find the dryness fraction of this steam, taking the specific volume of dry saturated steam at 88 lb. per square inch absolute pressure as 5 cubic feet.

8. The temperature of 80 lb. of water is to be raised from 10° C. (50° F.) to 90° C. (194° F.) by passing steam into it. The steam has an absolute pressure of 30 lb. per square inch and its dryness fraction is 0·96. What is the minimum weight of steam required?

9. Having given that for saturated steam at a pressure of 100 lb. per square inch the specific volume is 4·44 cubic feet, total heat 660·4 C.H.U. (1188·8 B.Th.U.), and latent heat 494·7 C.H.U. (890·5 B.Th.U.), what is the internal energy of the dry steam and of the steam when its dryness fraction is 0·9?

10. If the entropy of water at 160° C. is 0·4639 what is the entropy of steam at the same temperature whose dryness fraction is 0·85? Take the latent heat of the dry steam as 498·1 C.H.U.

11. Assuming the specific heat of water to be constant and equal to unity the

formula $\phi_w = \log_e(273 + t) - \log_e 273$ gives the entropy of water at t° C. Use this formula to calculate the entropy of water at 20° C., 60° C., 100° C., and 140° C.

Also, given that the latent heats of dry saturated steam at the above temperatures are, 584·5, 562·6, 539·2, and 512·7 C.H.U. respectively, calculate the entropy ϕ_s of dry saturated steam at these temperatures. Plot the results to the following scales:—Temperature t, 1 cm. = 10°. Entropy, 10 cm. = 1 unit of entropy.

12. Given that for steam having a temperature of 180° C. the sensible heat is 182·1 C.H.U. and the latent heat 482·3 C.H.U. what must be the quality of steam of this temperature if its total heat is 640 C.H.U.?

13. Taking the necessary data from the saturated steam table calculate the dryness fraction of steam at 180° C., 160° C., 140° C., 120° C., and 100° C., which at each of these temperatures has a total heat of 600 C.H.U.

Plot the water and saturation lines of a temperature-entropy chart between the temperatures 100° C. and 180° C. and construct the constant total heat curve representing 600 C.H.U. from the dryness fractions asked for in the first part of this exercise.

14. Steam expands isentropically from a temperature of 180° C. to a temperature of 100° C. If the dryness fraction is the same at the end as at the beginning of the expansion, calculate this dryness fraction and find under these conditions the dryness fraction at the intermediate temperature 140° C.

15. One pound of steam expands adiabatically from a pressure of 100 lb. per sq. in. to a pressure of 20 lb. per sq. in. The dryness fraction of the steam at the beginning of the expansion is 0·95. Find the volume of the steam at the end of the expansion and, assuming that pv^n = constant, determine the value of n.

16. Steam at a pressure of 180 lb. per sq. in. has 100° C. of superheat. This steam expands adiabatically until it becomes just saturated; what will then be its pressure? If pv^n = constant, is true for this expansion, find the value of n.

17. Steam is throttled from a pressure of 160 lb. per sq. in. to a pressure of 20 lb. per sq in. If the steam is dry and saturated at the end of the expansion, what is its dryness fraction at the beginning? By how much is the entropy of the steam increased by throttling?

18. In a test with a throttling calorimeter the pressure of the steam in the steam pipe was 130 lb. per sq. in. absolute. The temperature and pressure after throttling were 130° C. (266° F.) and 15 lb. per sq. in. absolute. What was the dryness fraction of the steam in the steam pipe?

19. What is the minimum dryness fraction which may be determined by a throttling calorimeter if the steam to be tested has an absolute pressure of 200 lb. per sq. in. and the absolute pressure of the steam after throttling is 15 lb. per sq in.?

20. A throttling calorimeter is supplied with dry saturated steam at a pressure of 100 lb. per sq. in. absolute. The exhaust pipe of the calorimeter is provided with a valve the opening of which is adjusted to make the pressure in the calorimeter 20 lb. per sq. in. absolute. The temperature of the steam in the calorimeter after throttling is then found to be 145° C. (293° F.). From these particulars, and the data obtainable from a saturated steam table, compute the mean specific heat of superheated steam at 20 lb. per sq. in. pressure between the temperature of saturation and 145° C. (293° F.).

21. Steam on its way to an engine passes through a separator in which the greater part of the water in the steam is deposited. In one hour the weight of water collected in the separator is 11·2 lb. and the engine uses 212 lb. of steam in that time. Tested with a throttling calorimeter, the steam leaving the separator is found to have a dryness fraction of 0·99. Calculate the dryness fraction of the steam entering the separator.

22. Steam at an absolute pressure of 90 lb. per sq. in. is passed into a tank containing water where it is condensed. Before the steam is introduced the weight of water in the tank is 148 lb. The "water equivalent" of the tank is 2 lb., that is, the capacity for heat of the tank is the same as that of 2 lb. of water. The temperature of the water in the tank before the steam is introduced is 15° C. (59° F.), and after, 35° C. (95° F.). By weighing the tank and its contents before and after the steam is introduced it is found that the weight of steam condensed is 5 lb. Compute from the above data, and data from a saturated steam table, the dryness fraction of the steam.

CHAPTER IV

IDEAL HEAT ENGINE CYCLES

64. The Elementary Heat Engine.—Since a heat engine is an instrument for converting heat into work the first requisite for such an engine, apart from its mechanism, is evidently a *supply of heat*. Excepting the few heat engines which have used directly the heat of the sun, heat engines have hitherto used the heat produced by the combustion of fuel. The second requisite is a *working substance* which while taking in heat undergoes change of pressure or change of volume or change of both pressure and volume. Hitherto the working substance has always been a fluid. The working fluid by increasing its volume while it is absorbing heat or after it has absorbed heat is made to do external work under the direction of the mechanism of the engine and this work is done at the expense of the heat conveyed to the working fluid from the source of heat. After the external work has been done the working fluid which has been used must either be discharged to waste with the heat remaining in it and a fresh supply made use of to continue the operation of the engine, or the working fluid must be cooled and compressed or condensed so as to bring it back to its original state in order that it may again take up heat from the source and again be used in the engine.

In studying the theory of heat engines it is most convenient to assume that the working fluid is used over and over again in the engine, in which case the fluid passes through a cycle of changes, the cycle being repeated over and over again. The cycle of changes or cycle of operations through which the working fluid passes in a heat engine is called the *engine cycle*. In studying the theory of a heat engine an ideal cycle is assumed and this cycle is carried out in an ideal way so as to give the best result and the engine is a *perfect heat engine*.

Although the ideal performance of a perfect heat engine may be far from possible with an actual engine it is instructive to determine the ideal performance to form a standard of comparison for actual engines.

In studying the various heat engine cycles it is usual to assume that 1 lb. weight of the working fluid is in use.

65. Carnot Cycle.—Sadi Carnot (*b.* 1796, *d.* 1832), a French physicist, was the first to study the performance of an ideal heat engine in which the cycle consisted of four operations, two being isothermal and two adiabatic, and this cycle, known as the *Carnot cycle*, will now be examined.

The essential parts of the ideal engine working under the Carnot cycle are—(*a*) a cylinder and piston (Figs. 68 to 71) made of material which is a perfect non-conductor of heat and having a fixed cover CC which is a perfect conductor of heat; (*b*) a movable cover NC which is a perfect non-conductor of heat; (*c*) a movable hot body HB kept at a constant absolute temperature T_1; (*d*) a movable cold body CB kept at a constant absolute temperature T_2. The non-conducting cover NC, the hot body HB, and the cold body CB may be placed separately against the conducting cover CC.

For the present the working fluid will be assumed to be a perfect gas.

First operation. (Fig. 68.)—At the beginning of the first operation there is in the cylinder behind the piston a volume V_1 of gas at a pressure P_1 and absolute temperature T_1. The hot body HB is in

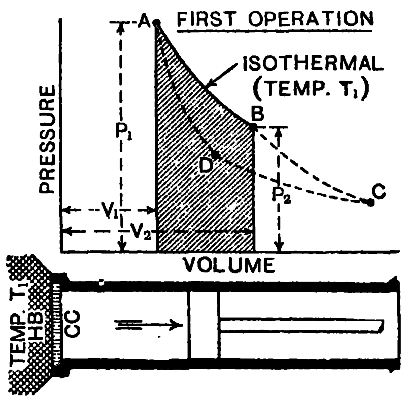


Fig. 68.

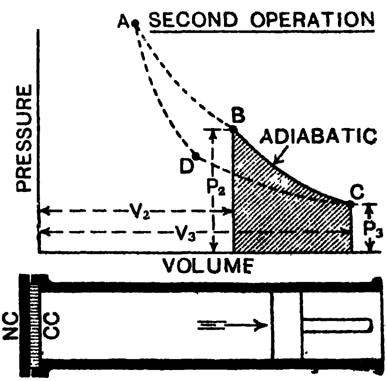


Fig. 69.

position as shown and the gas expands at constant temperature T_1 until its volume becomes V_2 and pressure P_2. The work done is represented by the shaded area and is equal to

$$P_1V_1 \log_e \frac{V_2}{V_1} = RT_1 \log_e \frac{V_2}{V_1}.$$

The heat equivalent of this work is the amount of heat given to the gas by HB, since the internal energy of the gas is unchanged.

At the end of the first operation HB is removed and its place taken by NC.

Second operation. (Fig. 69.)—The gas expands adiabatically until its volume is V_3, its pressure falling to P_3 and its temperature to T_2.

The work done is represented by the shaded area and is equal to $\dfrac{P_2V_2 - P_3V_3}{\gamma - 1}$.

This work is done at the expense of the internal energy of the gas.

At the end of the second operation NC is removed and its place taken by CB.

Third operation. (Fig. 70.)—The gas is compressed at the constant temperature T_2 until its volume is V_4, its pressure rising to P_4.

The work done is represented by the shaded area and is equal to $P_4V_4 \log_e \frac{V_3}{V_4} = RT_2 \log_e \frac{V_3}{V_4}$.

The heat equivalent of this work is the amount of heat transferred from the gas to the cold body CB, since the internal energy of the gas is unchanged.

At the end of the third operation CB is removed and its place taken by NC.

Fourth operation. (Fig. 71.)—The gas is compressed adiabatically until its volume is V_1, its pressure rising to P_1 and temperature to T_1.

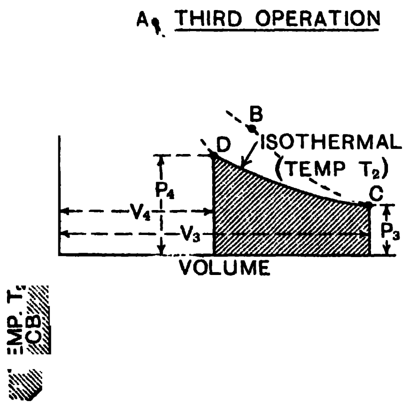


FIG. 70.

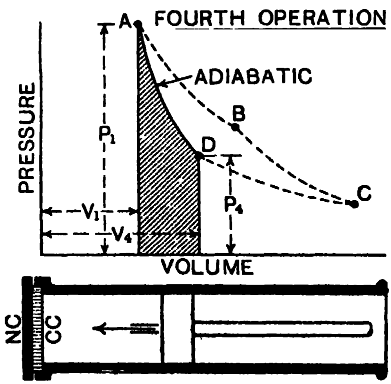


FIG. 71.

The work done is represented by the shaded area and is equal to $\frac{P_1V_1 - P_4V_4}{\gamma - 1}$.

The heat equivalent of this work goes to increase the internal energy of the gas.

In order that the volume, pressure, and temperature of the gas at the end of the fourth operation may be the same as at the beginning of the first operation there must be a certain relation between the volumes V_1, V_2, V_3, and V_4 which is determined as follows.

For the adiabatic expansion $\frac{T_1}{T_2} = \left(\frac{V_3}{V_2}\right)^{\gamma-1}$

For the adiabatic compression $\frac{T_1}{T_2} = \left(\frac{V_4}{V_1}\right)^{\gamma-1}$

Hence $\frac{V_3}{V} = \frac{V_4}{V_1}$ and $\frac{V_2}{V_1} = \frac{V_3}{V} = r$

That is to say, the ratio of isothermal expansion in the first operation is the same as the ratio of isothermal compression in the third operation.

Since $P_1V_1 = P_2V_2$ and $P_3V_3 = P_4V_4$ it follows that the work

done during adiabatic expansion, namely, $\dfrac{P_2V_2 - P_3V_3}{\gamma - 1}$ is equal to the work done during adiabatic compression, namely, $\dfrac{P_1V_1 - P_4V_4}{\gamma - 1}$, and the net work done during a cycle is represented by the shaded area ABCD, Fig. 72, and is equal to $P_1V_1 \log_e r - P_1V_4 \log_e r = (P_1V_1 - P_4V_4) \log_e r = R(T_1 - T_2) \log_e r$.

In one cycle the heat received from the hot body is $P_1V_1 \log_e r = RT_1 \log_e r$; the heat rejected to the cold body is $P_4V_4 \log_e r = RT_2 \log_e r$; and the heat converted into work is $R(T_1 - T_2) \log_e r$. Hence the *efficiency* of the engine as an instrument for converting heat into work is $\dfrac{R(T_1 - T_2) \log_e r}{RT_1 \log_e r} = \dfrac{T_1 - T_2}{T_1}$.

This result is more simply obtained by making use of the temperature-entropy diagram for the cycle shown in Fig. 73. The four

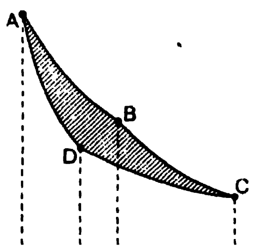


FIG. 72.

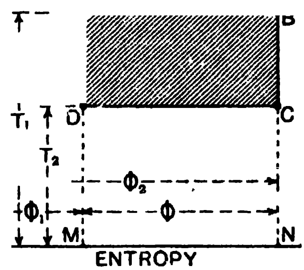


FIG. 73.

operations of the cycle are represented by the straight lines AB, BC, CD, and DA. The heat taken in during the first operation is represented by the area of the rectangle ABNM and is equal to $T_1(\phi_2 - \phi_1) = T_1\phi$. The heat rejected during the third operation is represented by the area of the rectangle CDMN and is equal to $T_2(\phi_2 - \phi_1) = T_2\phi$. The heat converted into work is represented by the area of the shaded rectangle ABCD and is equal to $(T_1 - T_2)\phi$, and the efficiency is

$$\frac{(T_1 - T_2)\phi}{T_1\phi} = \frac{T_1 - T_2}{T_1} \text{ as before.}$$

66. Carnot Cycle with Steam.—Using steam, or other vapour, as the working fluid, the isothermals AB and CD in the Figs. of the preceding Art. become also lines of constant pressure and the diagram of the cycle will be as in Fig. 74.

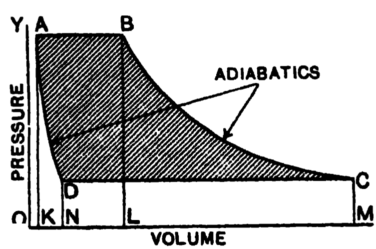


FIG. 74.

Starting at the point A, the four operations of the cycle are:

(1). A volume OK of water at pressure AK and temperature T_1 is heated at constant temperature. The water gradually evaporates and

when the evaporation is complete the steam produced will occupy a volume OL. This operation is represented by the horizontal line AB.

(2). The supply of heat being stopped the steam expands adiabatically until its volume is OM. Its pressure is reduced to CM and its temperature falls to T_2. This operation is represented by the adiabatic BC.

(3). The steam is compressed isothermally at temperature T_2 and it condenses as the heat is rejected to the cold body, but the cold body is removed before all the steam is condensed, the volume being then ON. This operation is represented by the horizontal line CD.

(4). The volume ON is so chosen that the adiabatic through D passes through A. The condition for this is that $\frac{OM}{ON} = \frac{OL}{OK}$. The contents of the cylinder are compressed adiabatically and the volume is reduced to OK, the pressure is raised to AK, and the temperature is raised to T_1. This operation is represented by DA.

The temperature entropy diagram is still a rectangle as in Fig. 73 and the efficiency is $\frac{T_1 - T_2}{T_1}$ as for a perfect gas.

The volume OK of the water is so small compared with the volume OL of the steam that it has had to be greatly exaggerated in Fig. 74 to make a clear diagram. If drawn to scale the adiabatic DA would practically lie on the axis OY.

67. Otto and Diesel Cycles.—These cycles are very important in connection with internal combustion engines and they are fully considered in Chapters XXII. to XXIV. It will be convenient, however, to state here that in the Otto cycle the heat is taken in at one constant volume and rejected at another constant volume, while in the Diesel cycle the heat is taken in at constant pressure and rejected at constant volume. In both of these cycles the expansion and compression curves are adiabatics.

68. Stirling Cycle.—A hot air engine, invented by the Rev. Robert Stirling in 1816, worked on a cycle, now known as the *Stirling cycle*, which in its ideal form is represented by the diagram ABCD in Fig. 75. There are four operations in a complete cycle:—

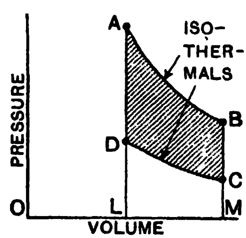


FIG. 75.

(1). Air having a volume $OL = V_1$, a pressure $AL = P_1$, and an absolute temperature T_1 expands isothermally until its volume is $OM = V_2$ and pressure $BM = P_2$. This operation is represented by the isothermal AB.

The work done is represented by the area ABML which is equal to $P_1V_1 \log_e \frac{V_2}{V_1} = RT_1 \log_e r$, and an amount of heat equivalent to this work is *taken in*.

(2). The air then passes through a *regenerator*, its volume remaining constant, and its temperature is lowered to T_2 and its pressure falls to $CM = P_3$. This operation is represented by the vertical line BC.

[The regenerator consists of a nest of thin metal plates with narrow passages between. The hot air passing in one direction through these

passages gives up heat to the metal plates. This heat is returned to the air in operation (4) when it flows in the opposite direction.]

The amount of heat given by the air to the regenerator is $k_v(T_1 - T_2)$.

(3). The air is next compressed isothermally at temperature T_2 until its volume is $OL = V_1$, and its pressure $DL = P_4$. This operation is represented by the isothermal CD.

The work done is represented by the area CDLM which is equal to $P_3V_3 \log_e \frac{V_2}{V_1} = RT_2 \log_e r$, and an amount of heat equivalent to this work is *rejected*.

(4). The air is restored to its initial condition in operation (1) by returning through the regenerator where it picks up the heat left there in operation (2). Its temperature is raised to T_1 and its pressure to P_1. This operation is represented by the vertical line DA.

The heat taken in is $RT_1 \log_e r$.

The heat rejected is $RT_2 \log_e r$.

The heat converted into work is $R(T_1 - T_2) \log_e r$.

$$\text{The efficiency is } \frac{R(T_1 - T_2)\log_e r}{RT_1 \log_e r} = \frac{T_1 - T_2}{T_1}$$
$$= \frac{\text{Shaded area ABCD}}{\text{Area ABML}}.$$

This efficiency is the same as that of the Carnot cycle.

For information on the construction and performance of the Stirling engine the student may refer to Rankine's "Steam Engine."

69. Joule Cycle.—This cycle was proposed by Joule and in its ideal form is represented by the diagram ABCD in Fig. 76. There are four operations in the complete cycle:—

(1) Air having a volume $OK = V_1$, a pressure $AK = P_1$, and a temperature T_1, is heated at constant pressure until its volume becomes $OL = V_2$, and its temperature is raised to T_2. This operation is represented by the horizontal line AB.

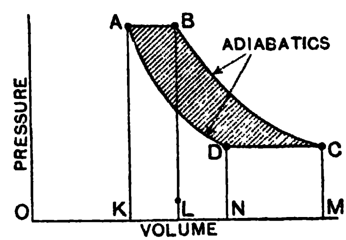


FIG. 76.

The heat received is $k_p(T_2 - T_1)$.

(2). The air then expands adiabatically until its volume is $OM = V_3$. Its pressure falls to $CM = P_3$ and its temperature is reduced to T_3. This operation is represented by the adiabatic BC.

No heat is received or rejected.

(3). The air is next cooled at constant pressure until its volume becomes $ON = V_4$. Its temperature is reduced to T_4. This operation is represented by the horizontal line CD.

The heat rejected is $k_p(T_3 - T_4)$.

(4). The air is restored to its initial condition in operation (1) by being compressed adiabatically. This operation is represented by the adiabatic DA.

No heat is received or rejected.

The heat converted into work is equal to the heat received minus the heat rejected which is $k_p(T_2 - T_1) - k_p(T_3 - T_4)$.

$$\text{Efficiency} = \frac{k_p(T_2 - T_1) - k_p(T_3 - T_4)}{k_p(T_2 - T_1)} = 1 - \frac{T_3 - T_4}{T_2 - T_1}.$$

$$\frac{T_3}{T_2} = \frac{P_3V_3}{P_2V_2}$$

But $P_3V_3^{\gamma} = P_2V_2^{\gamma}$, therefore $\frac{P_3V_3}{P_2V_2} = \left(\frac{V_2}{V_3}\right)^{\gamma-1} = \left(\frac{P_3}{P_2}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_3}{T_2}$

In like manner $\qquad \frac{T_4}{T_1} = \left(\frac{V_1}{V_4}\right)^{\gamma-1} = \left(\frac{P_4}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$

But $\qquad P_2 = P_1$ and $P_3 = P_4$

Hence, $\qquad \frac{V_2}{V_3} = \frac{V_1}{V_4} = \frac{1}{r}$ and $\frac{T_3}{T_2} = \frac{T_4}{T_1} = \left(\frac{1}{r}\right)^{\gamma-1}$

Now by a well known theorem in algebra, if $\frac{T_3}{T_2} = \frac{T_4}{T_1}$ each of these is equal to $\frac{T_3 - T_4}{T_2 - T_1}$

$$\text{Hence, efficiency} = 1 - \frac{T_3}{T_2} = \frac{T_2 - T_3}{T_2} = 1 - \left(\frac{1}{r}\right)^{\gamma-1}$$

$$= 1 - \frac{T_4}{T_1} = \frac{T_1 - T_4}{T_1} = \frac{\text{Shaded area ABCD}}{\text{Area ABCMK}}$$

The upper limit of temperature is T_2 and the lower limit is T_4 and the efficiency of the Carnot cycle for these limits would be $\frac{T_2 - T_4}{T_2}$ which is greater than $\frac{T_2 - T_3}{T_2}$ since T_4 is less than T_3. Hence the efficiency of the Joule cycle is less than that of the Carnot cycle for the same limits of temperature.

The Joule cycle is of interest because of its similarity to the cycles in modern internal combustion engines in which the heat is applied by combustion inside the cylinder, but Joule proposed to heat the air from an external source.

A reversed Joule cycle is used in the Bell-Coleman refrigerating machine.

70. Ericsson Cycle. –John Ericsson (b. 1803—d. 1889), a Swedish-American engineer, and a prolific inventor, made many attempts to produce a satisfactory hot air engine; he even had a ship built and fitted with his engines. The Ericsson cycle in its ideal form is represented by the diagram ABCD in Fig. 77. The four operations of the complete cycle are:—

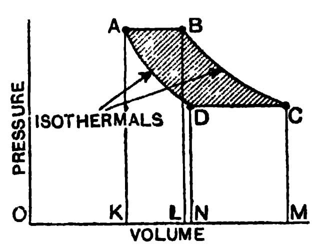


FIG. 77.

(1). Air having a volume $OK = V_1$, a pressure $AK = P_1$, and a temperature T_1 drawn, at constant pressure, through a regenerator in which its temperature is raised to T_2 and

its volume increased to $OL = V_2$. This operation is represented by the horizontal line AB.

The heat received from the regenerator is $k_p(T_2 - T_1)$.

(2). The air then expands isothermally at the temperature T_2 until its volume is $OM = V_3$, and its pressure falls to $CM = P_3$. This operation is represented by the isothermal BC.

The heat received is $RT_2 \log_e \frac{V_3}{V_2} = RT_2 \log_e r$.

(3). The air is next returned through the regenerator, at constant pressure, and gives up the heat which it received in operation (1). Its volume is reduced to $ON = V_4$ and its temperature falls to $T_4 = T_1$. This operation is represented by the horizontal line CD.

The heat returned to the regenerator is $k_p(T_2 - T_1)$.

(4). The air is compressed isothermally at the temperature T_1 until it reaches its initial state in operation (1). This operation is represented by the isothermal DA.

[In order that the isothermal rising from D shall pass through A it is easy to show that $\frac{V_3}{V_4} = \frac{V_2}{V_1}$ and $\frac{V_3}{V_2} = \frac{V_4}{V_1} = r$.]

The heat rejected is $RT_1 \log_e \frac{V_4}{V_1} = RT_1 \log_e r$.

Heat supplied $= RT_2 \log_e r$.

Heat converted into work $= RT_2 \log_e r - RT_1 \log_e r$.

$$\text{Efficiency} = \frac{R \log_e r(T_2 - T_1)}{RT_2 \log_e r} = \frac{T_2 - T_1}{T_2} = \frac{\text{Area ABCD}}{\text{Area ABCMK}}$$ which is the same as in the Carnot cycle.

71. Rankine Cycle.—The ideal steam engine cycle which is now generally used as the standard of comparison for actual steam engines, including steam turbines, is the *Rankine cycle* which is therefore of great importance. This cycle is a modified Carnot cycle.

Referring to Fig. 78, which is the pressure-volume diagram of the Rankine cycle, and comparing it with Fig. 74, p. 83, which is the pressure-volume diagram of the Carnot cycle when steam is the working fluid, the first and second operations, represented by AB and BC respectively, are the same in both cycles. In the third operation, represented by CD, the steam is completely condensed in the Rankine cycle while in the Carnot cycle the condensation is stopped before all the steam is condensed to enable the fourth operation, adiabatic compression, to restore the working fluid to its initial state in the first operation as represented by the point A.

In the Rankine cycle the working fluid, now water, at the end of the third operation is restored to the condition represented by the point A by being heated in the boiler or it may be heated partly or entirely in a feed-water heater before it is pumped into the boiler where it is converted into steam. It is however convenient to imagine that all the operations take place in the cylinder of the engine.

The volume ON, Fig. 78, is the volume of the feed water and this is so small compared with the volume OL of the steam produced from

it (ON is greatly exaggerated in Fig. 78) that it is usual to neglect it. The pressure-volume diagram of the Rankine cycle is then as shown in Fig. 79. The corresponding temperature-entropy diagrams for three

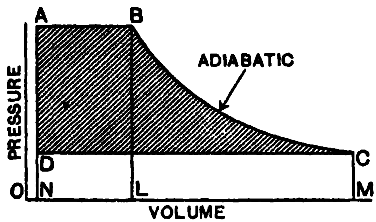


FIG. 78.

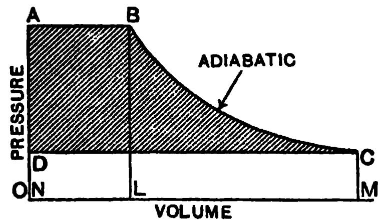


FIG. 79.

different conditions of the steam at the beginning of the adiabatic expansion are shown in Figs. 80, 81, and 82.

The four operations of the Rankine cycle will now be examined more fully and the efficiency of the cycle determined, reference being made to Figs. 79 to 82.

First operation. —Starting at the point A which represents 1 lb. of water at a pressure AN, temperature T_1 (absolute), which is the boiling temperature of water at pressure AN, and a volume which is neglected. The water is heated at constant pressure and it begins to

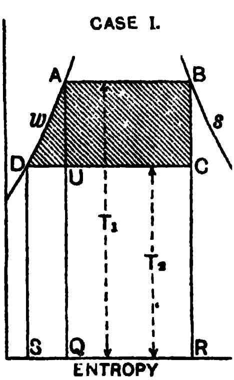


FIG. 80.

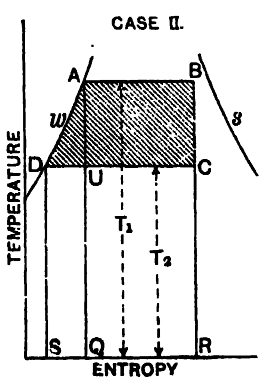


FIG. 81.

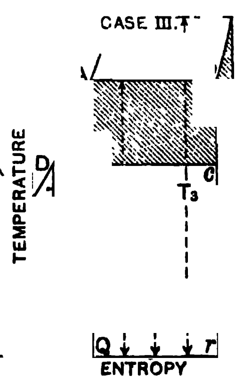


FIG. 82.

evaporate at once. The volume of steam increases until it reaches the value OL.

If this operation is stopped just when all the water is evaporated it is represented on the $T\phi$ diagram, Fig. 80, by the horizontal line AB, where A is on the water curve *w* and B is on the saturated steam curve *s*. The heat required to evaporate the water is L, the latent heat of steam at the pressure AN (Fig. 79), and this is represented by the area ABRQ (Fig. 80).

If this operation is stopped before all the water is evaporated, that is, if the steam is wet, the operation is represented on the $T\phi$

diagram, Fig. 81, by the horizontal line AB which ends before meeting the curve s. The heat added is xL, where x is the dryness fraction of the steam at B. This amount of heat is represented by the area ABRQ in Fig. 81.

If the first operation is continued after the water is all evaporated the steam becomes superheated, its temperature rises, but its pressure remains the same. This operation is now represented on the $T\phi$ diagram, Fig. 82, by the horizontal line Ab together with the curve bB. In the actual engine the evaporation would take place in the boiler and the superheating would be effected in a superheater. It may happen, however, that evaporation is not complete in the boiler, the steam leaving it being in a wet state, and in that case the evaporation is completed in the superheater. The heat added during the evaporation process is L and during the superheating process it is $k_p(T_3 - T_1)$, where T_3 is the temperature to which the steam is superheated and k_p is the mean specific heat of the steam at constant pressure between the temperatures T_1 and T_3. The whole heat added during the first operation is now represented on the $T\phi$ diagram, Fig. 82, by the area AbBRQ.

In each of the three cases represented by Figs. 80, 81, and 82, the heat received during the first operation is $H_B - h_A$ where H_B is the total heat of the steam at B and h_A is the heat of the water at A.

Second operation.—The steam expands adiabatically until its pressure and volume become CM and OM respectively (Fig. 79) and its temperature T_2. This operation is represented by the curve BC in Fig. 79 and by the vertical straight line BC in the corresponding $T\phi$ diagram.

At the end of the adiabatic expansion to C the total heat has been reduced to H_C which may be found from the total heat-entropy chart.

Third operation.—The steam is now compressed isothermally, the pressure remaining constant, and is completely condensed. This operation is represented by the horizontal line CD in Figs. 79 to 82. In the actual engine the condensation takes place in a condenser outside the engine cylinder. The heat rejected in this operation is $H_C - h_D$, where h_D is the heat of the water at D. This amount of heat is represented by the area CDSR in Figs. 80, 81, and 82.

Fourth operation.—The water resulting from the condensation of the steam in the third operation has its pressure raised to AN (Fig. 79) by being pumped into the boiler, and its temperature is raised to T_1 in the boiler, or partly or entirely in a feedwater heater before entering the boiler.

The heat required to raise the temperature of the water from T_2 to T_1 is $h_A - h_D$.

The heat equivalent of the work done in pumping the feed water into the boiler is so small that it is usually neglected.

Efficiency.—The heat received during the first and fourth operations is $H_B - h_A + (h_A - h_D) = H_B - h_D$.

The heat rejected during the third operation is $H_C - h_D$.

No heat is received or rejected during the second operation.

Heat converted into work $= H_B - h_D - (H_C - h_D) = H_B - H_C$.

$$\text{Efficiency} = \frac{H_B - H_C}{H_B - h_D} = \frac{\text{Area ABCD}}{\text{Area DABRS}}, \text{ Figs. 80 and 81,}$$

$$= \frac{\text{Area A}b\text{BCD}}{\text{Area DA}b\text{BRS}}, \text{ Fig. 82.}$$

This is the expression for the efficiency which is most convenient for use in practice. The values of H_B, H_C, and h_D are readily found from steam tables and the total heat-entropy chart.

When the efficiency of the Rankine cycle has to be calculated with the minimum use of steam tables or charts the formulæ now to be proved are used. The three cases illustrated by Figs. 80, 81, and 82, will be considered separately.

Case I. Fig. 80.—Steam dry and saturated at the beginning of the adiabatic expansion.

Area DAQS $= k(T_1 - T_2)$ where k is the mean specific heat of the water between the temperatures T_2 and T_1

The entropy SQ $= k \log_e \frac{T_1}{T_2}$. (Art. 55, p. 59.)

$$\text{Area UDSQ} = \text{SD} \times \text{SQ} = kT_2 \log_e \frac{T_1}{T_2}$$

$$\text{Area DAU} = k(T_1 - T_2) - kT_2 \log_e \frac{T_1}{T_2}$$

Area ABRQ $= L_1 =$ latent heat at temperature T_1

$$\text{Area CUQR} = L_1 \times \frac{T_2}{T_1}. \quad \text{Area ABCU} = L_1 - L_1 \times \frac{T_2}{T_1}$$

$$\text{Efficiency} = \frac{\text{Area DAU} + \text{area ABCU}}{\text{Area DAQS} + \text{area ABRQ}}$$

$$= \frac{k(T_1 - T_2) - kT_2 \log_e \frac{T_1}{T_2} + L_1 - L_1 \times \frac{T_2}{T_1}}{k(T_1 - T_2) + L_1}$$

$$= 1 - \frac{T_2\left(k \log_e \frac{T_1}{T_2} + \frac{L_1}{T_1}\right)}{k(T_1 - T_2) + L_1}$$

If k is taken $= 1$, which is commonly done, then,

$$\text{Efficiency} = 1 - \frac{T_2\left(\log_e \frac{T_1}{T_2} + \frac{L_1}{T_1}\right)}{T_1 - T_2 + L_1}$$

Case II. Fig. 81.—Steam wet at the beginning of the adiabatic expansion. Dryness fraction $= x_1$.

The efficiency in this case is derived from that of Case I. by substituting x_1L_1 for L_1.

CASE III. Fig. 82.—Steam superheated to temperature T_3 at the beginning of the adiabatic expansion.

Areas DAQS and DAU same as in Case I.

Area $AbrQ$ same as area ABRQ in Case I.

Area $AbcU$ same as area ABCU in Case I.

Area $bBRr = k_p(T_3 - T_1)$, where k_p is the mean specific heat of steam at constant pressure between T_1 and T_3.

$$\text{Entropy } rR = k_p \log_e \frac{T_3}{T_1}$$

$$\text{Area } CcrR = rc \times rR = k_p T_2 \log_e \frac{T_3}{T_1}$$

$$\text{Area } bBCc = k_p(T_3 - T_1) - k_p T_2 \log_e \frac{T_3}{T_1}$$

$$\text{Efficiency} = \frac{\text{Area DAU} + \text{area } AbcU + \text{area } bBCc}{\text{Area DAQS} + \text{area } AbrQ + \text{area } bBRr}$$

$$= \frac{k(T_1 - T_2) - kT_2 \log_e \frac{T_1}{T_2} + L_1 - L_1 \times \frac{T_2}{T_1} + k_p(T_3 - T_1) - k_p T_2 \log_e \frac{T_3}{T_1}}{k(T_1 - T_2) + L_1 + k_p(T_3 - T_1)}$$

$$= 1 - \frac{T_2\left(k \log_e \frac{T_1}{T_2} + \frac{L_1}{T_1} + k_p \log_e \frac{T_3}{T_1}\right)}{k(T_1 - T_2) + L_1 + k_p(T_3 - T_1)}$$

Taking $k = 1$

$$\text{Efficiency} = 1 - \frac{T_2\left(\log_e \frac{T_1}{T_2} + \frac{L_1}{T_1} + k_p \log_e \frac{T_3}{T_1}\right)}{T_1 - T_2 + L_1 + k_p(T_3 - T_1)}$$

Steam consumption per horse-power per hour.—One horse-power for one hour $= 33000 \times 60$ ft.-lb. $= \frac{33000 \times 60}{J}$ heat units.

Work done per lb. of steam in Rankine engine $= H_B - H_C$.

Steam per horse-power per hour $= \frac{33000 \times 60}{J(H_B - H_C)}$ lb.

EXAMPLE.—The upper and lower pressures in a Rankine cycle are 180 and 3 lb. per square inch absolute respectively. It is required to find the efficiency of the cycle, and the steam consumption per horse-power per hour, when the steam at the beginning of the adiabatic expansion is—(*a*) dry and saturated, (*b*) wet, the dryness fraction being 0·95, and (*c*) superheated 100° C.

(*a*) This is Case I., Fig. 80. From the steam table $H_B = 666{\cdot}6$ C.H.U., and $h_D = 60{\cdot}7$ C.H.U. From the total heat-entropy chart $H_C = 514$ C.H.U.

Hence, efficiency $= \frac{H_B - H_C}{H_B - h_D} = \frac{666{\cdot}6 - 514}{666{\cdot}6 - 60{\cdot}7} = 0{\cdot}252$ or $25{\cdot}2$ per cent.

Using the formula, efficiency $= 1 - \dfrac{T_2\left(\log_e \dfrac{T_1}{T_2} + \dfrac{L_1}{T_1}\right)}{T_1 - T_2 + L_1}$

$T_1 = 273 + 189 \cdot 5 = 462 \cdot 5^\circ$ C. absolute.

$T_2 = 273 + 60 \cdot 8 = 333 \cdot 8^\circ$ C. absolute. $L_1 = 474 \cdot 5$ C.H.U.

$\log_e \dfrac{T_1}{T_2} = 2 \cdot 3026 \log_{10} \dfrac{T_1}{T_2} = 0 \cdot 3261.$ $\dfrac{L_1}{T_1} = 1 \cdot 0259.$

Efficiency $= 1 - \dfrac{333 \cdot 8(0 \cdot 3261 + 1 \cdot 0259)}{462 \cdot 5 - 333 \cdot 8 + 474 \cdot 5} = 0 \cdot 252$ or $25 \cdot 2$ per cent.

Steam consumption $= \dfrac{33000 \times 60}{1400(666 \cdot 6 - 514)} = 9 \cdot 27$ lb. per H.P. per hour.

(b) This is Case II., Fig. 81. $h_D = 60 \cdot 7$ C.H.U. as in (a). From a total heat-entropy chart $H_B = 643$ C.H.U. and $H_C = 497$ C.H.U.

Hence, efficiency $= \dfrac{643 - 497}{643 - 60 \cdot 7} = 0 \cdot 251$ or $25 \cdot 1$ per cent.

The same result is obtained by means of the formula—

$$\text{Efficiency} = 1 - \frac{T_2\left(\log_e \dfrac{T_1}{T_2} + \dfrac{x_1 L_1}{T_1}\right)}{T_1 - T_2 + x_1 L_1}$$

Steam consumption $= \dfrac{33000 \times 60}{1400(643 - 497)} = 9 \cdot 69$ lb. per H.P. per hour.

(c) This is Case III., Fig. 82 $h_D = 60 \cdot 7$. From the superheated steam table $H_B = 721 \cdot 8$ C.H.U., and from the total heat-entropy chart $H_C = 550$ C.H.U.

Hence, efficiency $= \dfrac{721 \cdot 8 - 550}{721 \cdot 8 - 60 \cdot 7} = 0 \cdot 260$ or $26 \cdot 0$ per cent.

The same result is obtained by means of the formula—

$$\text{Efficiency} = 1 - \frac{T_2\left(\log_e \dfrac{T_1}{T_2} + \dfrac{L_1}{T_1} + k_p \log_e \dfrac{T_3}{T_1}\right)}{T_1 - T_2 + L_1 + k_p(T_3 - T_1)},\ k_p \text{ being} = 0 \cdot 55.$$

Steam consumption $= \dfrac{33000 \times 60}{1400(721 \cdot 8 - 550)} = 8 \cdot 23$ lb. per H.P. per hour.

72. Some Terms and Theorems from Thermodynamics.—In works on thermodynamics, the science of the relations between heat and work, there are certain terms used which may be stated here, also some theorems.

A *reversible process* is one which may be performed backwards, the state of the substance upon which the process is performed being the

same at the end of the backward operation as at the beginning of the forward operation. Adiabatic and isothermal changes of pressure and volume of a perfect gas are examples of reversible processes. The process of producing heat by friction is an irreversible process.

If all the operations of the cycle of an engine are reversible the cycle is a *reversible cycle* and the engine is a *reversible engine*. That is to say, if a reversible heat engine converts a certain amount of heat H into an amount of work W, and if this amount of work W be employed in driving the engine backwards, this work will be reconverted into an amount of heat H.

The *conditions for maximum efficiency* of a heat engine are—(1) All the heat received must be received at the higher temperature and all the heat rejected must be rejected at the lower temperature. (2) The drop in temperature betweeen the source of heat and the working substance, when heat is being received, must be indefinitely small, and the drop in temperature between the working substance and the condenser or cold body, when heat is being rejected, must be indefinitely small. (3) There must be no free or unbalanced expansion.

For the same temperature range *all reversible engines have the same efficiency* and the efficiency is independent of the working substance. This is one form of the statement of the *second law of thermodynamics*. Another form of the statement of this law is—a self-acting engine, unaided by external agency, cannot cause heat to pass from one body to another at a higher temperature. This is equivalent to the statement that heat cannot pass by conduction or radiation from one body to another at a higher temperature.

For a given temperature range a reversible heat engine has the maximum efficiency and this is equal to $\frac{T_1 - T_2}{T_1}$ as in the Carnot engine.

Exercises IV

1. A perfect heat engine works on the Carnot cycle between 1000° C. and 200° C. If this engine receives heat at the higher temperature at the rate of 2000 C.H.U. per minute compute the horse-power of the engine.

2. Half a pound of air passes through a Carnot cycle between the temperatures 205° C. (401° F.) and 100·6° C. (213° F.). The ratio of isothermal expansion is 3, and the initial pressure of the air is 300 lb. per sq. inch absolute.

If $\gamma = 1{\cdot}409$, and if the specific heat of air at constant volume is 0·17, find:—

(*a*) The pressure and volume of the air at the end of each of the four stages.

(*b*) The thermal efficiency of the cycle. [U.L.]

3. Compute the efficiency of an ideal Stirling cycle in which the expansion ratio is 3 and the ratio of the maximum pressure to the minimum pressure is 6.

4. In a hot air engine the air takes in heat at constant volume, the temperature rising from 130° C. to 1000° C. The useful work done per lb. of air is 60,000 foot-pounds. Calculate the thermal efficiency, taking the specific heat of air at constant volume as 0·17.

5. Steam is generated at a pressure of 200 lb. per sq. inch absolute and is then superheated to a temperature of 350° C. (662° F.).

If this superheated steam is used in an engine working on the Rankine cycle, and if the lower limit of pressure is 1 lb. per sq. inch absolute, calculate:—

(*a*) The thermal efficiency.

(b) The dryness fraction of the steam at the end of the expansion.

Pressure.	Temperature.		Latent Heat.		Liquid	Evaporation
lb.	C.	F.	C.H.U.	B.Th.U.	Entropy.	Entropy $= \frac{L}{T}$
200	194·4°	381·9°	472·2	849·9	0·5437	1·0101
1	38·7°	101·6°	573·8	1032·8	0·1323	1·8401

The mean specific heat of superheated steam between 194·4° C. and 350° C. may be taken as 0·52. [U.L.]

6. Compare the efficiencies of two engines working on the Rankine cycle, one uses saturated steam at a pressure of 110 lb. per sq. inch absolute, and the other, superheated steam at the same pressure at a temperature of 250° C. (482° F.). The exhaust temperature in both cases is 55° C. (131° F.). The mean specific heat of superheated steam for above range may be taken as 0·54. [U.L.]

CHAPTER V

COMBUSTION AND FUEL

73. Chemical Principles.—Chemically considered all matter may be divided into two classes, *elements* and *compounds*. Water, for example, is a compound which may be decomposed into the elements oxygen and hydrogen. The substances known as elements are those which, so far, have resisted all attempts to divide them into parts which have different properties. It is necessary to distinguish between a chemical compound and a mere mixture of two or more elements or compounds. Two or more substances may be *mixed* together in any proportions and the mixture will partake of the properties of the separate constituents, but when substances combine to form a chemical compound the substances must be in certain definite proportions, and the properties of the compound are generally very different from those of its constituents.

The *atomic theory* of matter is of great assistance in comprehending chemical combinations and decompositions. In this theory it is supposed that an element is made up of extremely small particles which never subdivide and which are called *atoms*. All the atoms of a particular element are exactly alike, but the atoms of different elements are unlike. When one element combines with another to form a chemical compound, a certain definite number of atoms of the one unite with a certain definite number of atoms of the other to form a *molecule* of the chemical compound. A *molecule* may be defined as the smallest portion of a substance which can have a separate existence.

In chemistry the atoms of the various elements are denoted by letters thus: hydrogen H, oxygen O, nitrogen N, carbon C, and sulphur S, which are the elements which have to be considered in studying fuel and its combustion.

A suffix to a symbol which represents an atom, denotes a number of atoms. Thus, N represents one atom of nitrogen, while N_3 denotes three atoms of nitrogen. A chemical compound is represented by placing together the symbols which represent the atoms forming a molecule of the compound. Thus, CO_2 represents carbon dioxide, a compound of carbon and oxygen, the molecule consisting of 1 atom of carbon and 2 atoms of oxygen. A number placed in front of the formula for a molecule denotes that number of molecules. Thus, $2CO_2$ denotes 2 molecules of carbon dioxide.

Combustion is the chemical combination which takes place between the constituents of a fuel and oxygen when the fuel burns.

The combustion of hydrogen will first be considered. Two atoms of hydrogen combine with one atom of oxygen to form one molecule of steam, which will generally condense to water. This combination is represented by the equation—

$$\begin{array}{ccccc} \text{Hydrogen} & & \text{Oxygen} & & \text{Water} \\ H_2 & + & O & = & H_2O \end{array}$$

A molecule of oxygen consists of two atoms, so does a molecule of hydrogen, and the equation which represents the combustion of hydrogen is sometimes written: $2H_2 + O_2 = 2H_2O$, which means that two molecules of hydrogen combine with one molecule of oxygen to form two molecules of steam or water.

A most important law may now be stated. According to the kinetic theory of gases, and also from purely chemical considerations, *equal volumes of different gases at the same pressure and temperature contain the same number of molecules.* The temperature must however be sufficiently above that at which the gas liquefies. For example, if a cubic foot of hydrogen gas contains n molecules then a cubic foot of oxygen gas and a cubic foot of steam gas will each contain n molecules, the gases being at the same pressure and temperature. The equation $2H_2 + O_2 = 2H_2O$ may be written thus : $2(nH_2) + (nO_2) = 2(nH_2O)$, which shows that 2 cubic feet of hydrogen gas combine with 1 cubic foot of oxygen gas to form 2 cubic feet of steam gas, all at the same pressure and temperature.

Since equal volumes of gases contain the same number of molecules it follows that the relative weights of equal volumes of gases will be the same as the relative weights of their molecules.

The relative weights of the atoms or the *atomic weights* of the elements are expressed in terms of the weight of an atom of hydrogen. Thus, H = 1, O = 16, N = 14, C = 12, and S = 32.

The following table gives the atomic weights of the elements and the molecular weights of the gaseous elements and chemical compounds which are required in combustion calculations.

Element or Compound.	Atomic Weight.	Molecular Weight.	
Hydrogen	H = 1	$H_2 = 1 \times 2$	= 2
Oxygen	O = 16	$O_2 = 16 \times 2$	= 32
Nitrogen	N = 14	$N_2 = 14 \times 2$	= 28
Carbon	C = 12		
Sulphur	S = 32		
Water or steam		$H_2O = 1 \times 2 + 16$	= 18
Carbon dioxide		$CO_2 = 12 + 16 \times 2$	= 44
Carbon monoxide		$CO = 12 + 16$	= 28
Methane or Marsh gas		$CH_4 = 12 + 1 \times 4$	= 16
Ethylene or Olefiant gas		$C_2H_4 = 12 \times 2 + 1 \times 4$	= 28
Acetylene		$C_2H_2 = 12 \times 2 + 1 \times 2$	= 26
Sulphur dioxide		$SO_2 = 32 + 16 \times 2$	= 64

From the atomic weights of the elements the relative *weights* of the elements and compounds which enter into chemical combination

with one another or which result from such combination are determined, and from the molcular weights the relative *volumes* of the gases entering into chemical combination or resulting from such combination are readily found.

The relative weights of the substances in the equation $2H_2 + O_2 = 2H_2O$ may now be written under them as follows :—

		$2H_2$	+	O_2	=	$2H_2O$
Relative weights		$2 \times 1 \times 2$	+	16×2	=	$2(1 \times 2 + 16)$
	that is	4	+	32	=	36
	or	1	+	8	=	9

which shows that 1 lb. of hydrogen combines with 8 lb. of oxygen to form 9 lb. of water.

Considering next the combustion of carbon : one atom of carbon combines with two atoms of oxygen to form carbon dioxide gas and this is represented by the equation

		Carbon.		Oxygen.		Carbon dioxide.
		C	+	O_2	=	CO_2
Relative weights		12	+	16×2	=	$12 + 16 \times 2$
	that is	12	+	32	=	44
	or	1	+	2·67	=	3·67

which shows that 1 lb. of carbon combines with 2·67 lb. of oxygen to form 3·67 lb. of carbon dioxide. Also the volume of the O_2 on the left-hand side of the above equation will be the same as the volume of the CO_2 on the right-hand side when at the same pressure and temperature.

Carbon is said to be completely burned when it combines with oxygen to form carbon dioxide (CO_2). There is, however, another compound of carbon and oxygen, carbon monoxide (CO), which may be formed when the carbon is burned in a supply of oxygen which is insufficient to form carbon dioxide. The combustion of carbon to carbon monoxide is represented by the equation

		Carbon.		Oxygen.		Carbon monoxide.
		2C	+	O_2	=	2CO
Relative weights		2×12	+	16×2	=	$2(12 + 16)$
	that is	24	+	32	=	56
	or	1	+	1·33	=	2·33

which shows that 1 lb. of carbon combines with 1·33 lb. of oxygen to form 2·33 lb. of carbon monoxide. Also the volume of oxygen used is half the volume of the carbon monoxide produced, at the same pressure and temperature.

Carbon monoxide will combine with oxygen to form carbon dioxide as shown by the equation

		Carbon monoxide.		Oxygen.		Carbon dioxide.
		2CO	+	O_2	=	$2CO_2$
Relative weights		$2(12 + 16)$	+	16×2	=	$2(12 + 16 \times 2)$
	that is	56	+	32	=	88
	or	1	+	0·57	=	1·57

which shows that 1 lb. of carbon monoxide combines with 0·57 lb. of

oxygen to form 1·57 lb. of carbon dioxide. Also, two volumes of carbon monoxide combine with one volume of oxygen to form two volumes of carbon dioxide, at the same pressure and temperature. Carbon monoxide burns with a blue flame.

Another reaction which occurs in furnaces is the reduction of CO_2 to CO when the CO_2 formed in one part of the furnace passes over red-hot carbon, as shown by the equation $CO_2 + C = 2CO$.

Sulphur occurs in fuels to a small extent only. The equation which represents the combustion of sulphur is as follows :—

		Sulphur.		Oxygen.		Sulphur dioxide.
		S	+	O_2	=	SO_2
Relative weights		32	+	16×2	=	$32 + 16 \times 2$
	that is	32	+	32	=	64
	or	1	+	1	=	2

which shows that 1 lb. of sulphur combines with 1 lb. of oxygen to form 2 lb. of sulphur dioxide.

Consider lastly, the combustion of one of the hydrocarbons, say, ethylene or olefiant gas, which has the molecular formula C_2H_4. The combustion of this gas in oxygen is represented by the equation

	Ethylene.		Oxygen.		Carbon dioxide.		Steam.
	C_2H_4	+	$3O_2$	=	$2CO_2$	+	$2H_2O$
Relative weights	$(12\times2+1\times4)$	+	$3(16\times2)$	=	$2(12+16\times2)$	+	$2(1\times2+16)$
	that is 28	+	96	=	88	+	36
	or 1	+	3·43	=	3·14	+	1·29
Relative numbers of molecules or relative volumes	1	+	3	=	2	+	2

The above shows that 1 lb. of ethylene requires 3·43 lb. of oxygen to form 3·14 lb. of carbon dioxide and 1·29 lb. of steam gas.

Also, 1 cubic foot of ethylene requires 3 cubic feet of oxygen to form 2 cubic feet of carbon dioxide and 2 cubic feet of steam gas, all at the same pressure and temperature.

When the products of combustion are cooled the steam condenses to a negligible volume. Hence it follows that the volume of the dried products of combustion in the above example is equal to twice the volume of the ethylene gas burned, the volumes being measured at the same pressure and temperature.

74. Composition of Air.—For the purpose of combustion calculations the composition of atmospheric air may be taken as follows :—

	By weight.	By volume.
Oxygen . .	23 per cent. . .	21 per cent.
Nitrogen . .	77 ,, . .	79 ,,

From the above it follows that 1 lb. of oxygen is associated with $77/23 = 3·35$ lb. of nitrogen in 4·35 lb. of air.

Also, 1 cubic foot of oxygen is associated with $79/21 = 3·76$ cubic feet of nitrogen in 4·76 cubic feet of air.

75. Composition of Fuels.—The subject of fuel will be treated

more fully further on in this chapter, but it will be convenient at this stage to state what are the constituents of fuels commonly used in connection with heat engines, and in what average proportions the constituents exist in the different fuels.

The important combustible constituents of fuels are carbon and hydrogen. Sulphur occurs to a small extent but its value as a combustible may in general be neglected. Besides these combustible constituents there are oxygen and nitrogen, and in solid fuels there are small quantities of mineral substances which form the ash when the fuel is burned.

When it is stated that a fuel consists of certain elements in given proportions it must not be understood that these elements exist separately in the fuel. The actual composition of most fuels is extremely complicated. The hydrogen, for example, will be in combination with the carbon forming, in general, a considerable number of different chemical compounds known as hydrocarbons. Chemical analysis will show what are the elements of which a fuel is composed and in what proportions these elements exist in the fuel, but the analysis does not generally show how these elements are combined together in the fuel because in the process of analysing it the various chemical compounds in the fuel are generally decomposed.

An analysis of a fuel into its elements does however enable the amount of air required for its combustion to be calculated. Also, from the analysis the heating value of the fuel may be calculated approximately.

The approximate average composition, per cent., by weight, of various fuels is given in the following table :—

Average Composition of Various Fuels.

Fuel.	Carbon.	Hydrogen.	Oxygen	Nitrogen.	Sulphur.	Ash.
Wood (dry) . . .	48·5	6·0	43·5	0·5	—	1·5
Peat (dry) . . .	58 0	6·3	30·8	0·9	—	4 0
Lignite (brown coal)	66·0	5·0	20·0	1·0	1·0	7·0
Bituminous coal .	81·0	5·0	8·0	1·5	1·0	3·5
Anthracite . . .	91·0	3·0	2·5	0·5	0·5	2·5
Petroleum (crude oil)	86·0	13·0	1·0	--	—	—

76. Air for Combustion—Products of Combustion.—The reactions which takes place when the elements of fuel are burned in oxygen have been given in Art. 73. The same reactions occur when these elements are burned in air, but the products of combustion will now contain the nitrogen of the air supplied. The nitrogen of the air undergoes no chemical change, that is, it does not combine with any of the elements of the fuel when the fuel is burned.

The various equations representing the complete combustion of the separate elements hydrogen, carbon and sulphur, in air will now be given showing the relative weights of the different substances and the relative volumes of the gases involved. Compounds and fuels of known composition will then be similarly dealt with.

Hydrogen.—1 lb. of hydrogen requires 8 lb. of oxygen to form 9 lb. of water or steam.

Hydrogen. Oxygen. Nitrogen. Water. Nitrogen.

$$2H_2 + O_2 \qquad\qquad = 2H_2O$$

Relative weights

$$1 + \underbrace{8 + 8 \times 3{\cdot}35}_{\text{air} = 8 \times 4{\cdot}35} = 9 + 8 \times 3{\cdot}35$$

$$1 + \underbrace{8 + 26{\cdot}8}_{\text{air} = 34{\cdot}8} = 9 + 26{\cdot}8$$

Relative volumes

$$2 + \underbrace{1 + 3{\cdot}76}_{\text{air} = 4{\cdot}76} \text{ become } 0 + 3{\cdot}76$$

The steam in the products of combustion is supposed to be condensed to water whose volume may be neglected. The dried products of combustion will consist of nitrogen only.

Carbon.—1 lb. of carbon requires 2·67 lb. of oxygen to form 3·67 lb. of carbon dioxide.

Carbon. Oxygen. Nitrogen. Carbon dioxide. Nitrogen.

$$C + O_2 \qquad\qquad = CO_2$$

Relative weights

$$1 + \underbrace{2{\cdot}67 + 2{\cdot}67 \times 3{\cdot}35}_{\text{air} = 2{\cdot}67 \times 4{\cdot}35} = 3{\cdot}67 + 2{\cdot}67 \times 3{\cdot}35$$

$$1 + \underbrace{2{\cdot}67 + 8{\cdot}94}_{\text{air} = 11{\cdot}61} = 3{\cdot}67 + 8{\cdot}94$$

Relative volumes

$$0 + \underbrace{1 + 3{\cdot}76}_{\text{air} = 4{\cdot}76} \text{ become } 1 + 3{\cdot}76$$

The carbon on the left hand side would be solid and its volume may be neglected.

The relative total weight of the products of combustion is 3·67 + 8·94 = 12·61. Hence the percentage composition of the products *by weight* is CO_2 29·1 and N 70·9.

The relative total volume of the products is 4·76. Hence the percentage composition of the products *by volume* is CO_2 21 and N 79.

Sulphur.—1 lb. of sulphur requires 1 lb. of oxygen to form 2 lb. of sulphur dioxide.

Sulphur. Oxygen. Nitrogen. Sulphur dioxide. Nitrogen.

$$S + O_2 \qquad\qquad = SO_2$$

Relative weights

$$1 + \underbrace{1 + 3{\cdot}35}_{\text{air} = 4{\cdot}35} = 2 + 3{\cdot}35$$

Relative volumes

$$0 + 1 + 3{\cdot}76 \text{ become } 1 + 3{\cdot}76$$

The percentage composition of the products *by weight* is SO_2 37·4 and N 62·6.

The percentage composition of the products *by volume* is SO_2 21 and N 79.

Oil fuel.—Taking the composition of an oil fuel, by weight, to be, carbon 85·1 per cent., hydrogen 13·7 per cent., and oxygen 1·2 per cent., the air required for the combustion of 1 lb. of this fuel and the composition of the dry products of combustion are determined as follows :—

The oxygen in 1 lb. of the fuel is sufficient for the combustion of $0·012 \div 8 = 0·0015$ lb. of hydrogen, leaving $0·137 - 0·0015 = 0·1355$ lb. of hydrogen to be provided with oxygen from the air.

Oxygen from air for hydrogen	$= 0·1355 \times 8$	$= 1·08$ lb.	
„ „ „ carbon	$= 0·851 \times 2·67$	$= 2·27$ lb.	
	Total	3·35 lb.	

Total air required per lb. of fuel $= 3·35 \times 4·35 = 14·57$ lb.
Nitrogen supplied per lb. of fuel $= 14·57 - 3·35 = 11·22$ lb.

Taking the volume of 1 lb. of air at 0° C. and 14·7 lb. per square inch pressure, as 12·39 cubic feet, the volume of air required per lb. of fuel $= 12·39 \times 14·57 = 180·5$ cubic feet at 0° C. and 14·7 lb. per square inch

The products of combustion, per lb. of fuel, will consist of

Carbon dioxide	$= 0·851 \times 3·67$	$= 3·12$ lb.
Water or steam	$= 0·137 \times 9$	$= 1·23$ lb.
Nitrogen		$= 11·22$ lb.
	Total	$= 15·57$ lb.

The *dry* products of combustion, per lb. of fuel, will contain

Carbon dioxide . . .	3·12 lb.
Nitrogen . . .	11·22 lb.
Total =	14·34 lb.

The percentage composition of the dry products by weight is therefore, carbon dioxide $= \frac{3·12 \times 100}{14·34} = 21·8$, and nitrogen $= \frac{11·22 \times 100}{14·34} = 78·2$

The molecular weights of hydrogen, carbon dioxide, and nitrogen are 2, 44, and 28 respectively. Let V be the volume of 1 lb. of hydrogen, then the volume of 3·12 lb. of CO_2 is $\frac{3·12 \times 2V}{44} = 0·142V$, and the volume of 11·22 lb. of N is $\frac{11·22 \times 2V}{28} = 0·801V$. Hence the volume of dry products per lb. of fuel is

$$0·142V + 0·801V = 0·943V.$$

The percentage composition of the dry products by volume is therefore, $CO_2 = \frac{0·142 \times 100}{0·943} = 15·1$, and $N = \frac{0·801 \times 100}{0·943} = 84·9$.

Coal.—The percentage composition of a Scotch bituminous coal, by weight, was found to be: C 76·6, H 5·3, O 12·8, N 2·1, S 1·2, and ash 2·0. It is required to find the minimum amount of air necessary for the complete combustion of 1 lb. of this coal and the percentage composition by weight and by volume of the dry gaseous products of combustion.

The oxygen in 1 lb. of this coal is sufficient for the combustion of $0·128 \div 8 = 0·016$ lb. of hydrogen, leaving $0·053 - 0·016 = 0·037$ lb. of hydrogen to be provided with oxygen from the air.

Oxygen from air for hydrogen	= $0·037 \times 8$	= 0·296 lb.
,, ,, ,, ,, carbon	= $0·766 \times 2·67$	= 2·045 ,,
,, ,, ,, ,, sulphur	= $0·012 \times 1$	= 0·012 ,,
	Total	= 2·353

Total air required per lb. of coal $= 2·353 \times 4·35 = 10·236$ lb.
Volume of air required per lb. of coal $= 10·236 \times 12·39$ 126·8 c. ft. at 0° C. and 14·7 lb. per square inch.
Nitrogen supplied per lb. of coal $= 10·236 - 2·353 = 7·883$ lb.
The products of combustion per lb. of coal will consist of

Water or steam	$0·053 \times 9$ =	0·477 lb.
Carbon dioxide	$0·766 \times 3·67$ =	2·811 ,,
Sulphur dioxide	$0·012 \times 2$ =	0·024 ,,
Nitrogen	$7·883 + 0·021$ =	7·904 ,,
Ash	=	0·020 ,,
	Total =	11·236

The *dry* gaseous products per lb. of coal are, CO_2 2·811, SO_2 0·024, and N 7·904, total 10·739 lb.

Expressed as percentages these become, CO_2 26·2, SO_2 0·2, N 73·6.

The molecular weights of H, CO_2, SO_2, and N are 2, 44, 64, and 28 respectively. If V is the volume of 1 lb. of hydrogen, then, the volumes of the gaseous constituents of the dry products per lb. of coal are: $CO_2 \frac{2·811 \times 2V}{44} = 0·128V$, $SO_2 \frac{0·024 \times 2V}{64} = 0·001V$, and $N \frac{7·904 \times 2V}{28} = 0·565V$, total $0·694V$.

The percentage composition of the dry gaseous products by volume is therefore, $CO_2 \frac{0·128 \times 100}{0·694} = 18·4$, $SO_2 \frac{0·001 \times 100}{0·694} = 0·1$, and $N \frac{0·565 \times 100}{0·694} = 81·4$.

Gaseous fuel.—The percentage composition of a producer gas by volume was—H 14, CH_4 1, CO 24, CO_2 5, O 1, and N 55. It is required to find the minimum volume of air necessary for the complete combustion of 100 cubic feet of this gas and the composition of the dry products of combustion by volume.

The equations representing the reactions which take place and the relative volumes of the gases involved when the combustibles H, CH_4, and CO are burned are as follows [1]

For hydrogen	$2H_2 + O_2 = 2H_2O$
Relative volumes	$2 + 1 = 2$
For methane	$CH_4 + 2O_2 = CO_2 + 2H_2O$
Relative volumes	$1 + 2 = 1 + 2$
For carbon monoxide	$2CO + O_2 = 2CO_2$
Relative volumes	$2 + 1 = 2$

Hence tabulating by volumes in cubic feet

	O		H_2O		CO_2		N
14 of hydrogen	+ 7	=	14		–		—
1 of methane	+ 2	=	2	+	1		—
24 of carbon monoxide	+ 12	=	—	+	24		—
5 of carbon dioxide	—	=	—		5		—
1 of oxygen	− 1	=	—		—		—
55 of nitrogen	—	=	—		—		55
100 cubic feet of gas	+ 20	=	16	+	30	+	55

Volume of air required $= 20 \times \frac{100}{21} = 95$ cubic feet.
Volume of nitrogen added with air $= 95 - 20 = 75$ cubic feet.
Volume of nitrogen in products $= 75 + 55 = 130$ cubic feet.

$$\text{Volume of dry products} = \overset{CO_2}{30} + \overset{N}{130} = 160 \text{ cubic feet.}$$

$$\text{,, ,, ,,} = 18{\cdot}75 + 81{\cdot}25 = 100 \text{ per cent.}$$

77. Determination of Air Supplied from Analysis of Flue Gases.—In the preceding Art. it has been shown how the minimum amount of air required for the complete combustion of a fuel of known composition may be calculated. But in order to ensure that each atom of the combustible part of the fuel shall meet its proper supply of atoms of oxygen, it is necessary to supply an excess of oxygen and therefore an excess of air. In actual practice the amount of this excess air depends on the character of the fuel, the kind of furnace, the method of stoking or of supplying the fuel to the furnace, and the pressure of the air supplied. Generally the excess air is from 50 per cent. to 100 per cent. of the theoretical minimum.

The amount of air required in ordinary furnaces is so great that it is not generally practicable to measure it directly, but it will now be shown how the amount of air actually passing through the furnace may be calculated from the analysis of the flue gases. The determination of the composition of the flue gases will not be described here, but it may be stated that the actual analysis is by volume and only the gases CO_2, CO, N, and O are generally determined. The SO_2 is usually

[1] In the equations connecting volumes before and after combustion, for "=" read "become."

so small as to be negligible and the steam is condensed but the resulting water is not generally measured.

The method of finding the air supplied will be best understood by considering an example.

Analysis of dry flue gases by volume : CO_2 11 per cent., CO 1 per cent., N 81 per cent., O 7 per cent. Also weight of carbon in fuel = 83 per cent.

Remembering that the molecular weights of gases are the relative weights of equal volumes of them, it follows that the relative weights of the given relative volumes of the gases in the above example are

$$\begin{array}{cccc} CO_2 & CO & N & O \\ 11 \times 44 : & 1 \times 28 : & 81 \times 28 : & 7 \times 32 \end{array}$$

The weight of carbon in 11×44 parts by weight of CO_2 $= \frac{12}{44} \times 11 \times 44 = 12 \times 11$.

The weight of carbon in 1×28 parts by weight of CO $= \frac{12}{28} \times 1 \times 28 = 12 \times 1$.

Total weight of carbon in the above parts of CO_2 and CO $= 12 \times 12$.

$$\frac{\text{Weight of nitrogen in flue gases}}{\text{Weight of carbon in flue gases}} = \frac{81 \times 28}{12 \times 12}.$$

But, neglecting the nitrogen in the fuel, this is also the ratio of the weight of nitrogen in the air supplied per pound of fuel to the weight of carbon in 1 lb of fuel. Hence, weight of nitrogen per pound of fuel $= \frac{81 \times 28}{12 \times 12} \times 0{\cdot}83$, and weight of air per pound of fuel $= \frac{81 \times 28}{12 \times 12} \times 0{\cdot}83 \times \frac{100}{77} = 17$ lb.

The excess air is determined from the amount of oxygen in the flue gases as follows:—Referring to the foregoing example, the 7 per cent. of oxygen is the oxygen of the excess air. The amount of nitrogen associated with this oxygen to form air is $7 \times \frac{79}{21} = 26{\cdot}3$ and the ratio of the excess air to the total air is $\frac{26{\cdot}3}{81}$. Therefore the excess air is $\frac{26{\cdot}3}{81} \times 17 = 5{\cdot}5$ lb., and the amount used for the combustion of the fuel is $17 - 5{\cdot}5 = 11{\cdot}5$ lb.

If C_2, C_1, N, and Y are the percentage volumes of CO_2, CO, N and O respectively in the dry flue gases, and if C is the percentage weight of carbon in the fuel, then, it is easy to show that,

$$\text{weight of air supplied per pound of fuel} = \frac{NC}{33(C_2 + C_1)}\text{lb.}$$

and, $$\text{weight of excess air per pound of fuel} = \frac{79YC}{21 \times 33(C_2 + C_1)}\text{ lb.}$$

Also, the excess air (by weight or volume) expressed as a percentage of the effective air used is given by the expression $\frac{100 \times 79Y}{21N - 79Y}$.

78. Calorific Value of a Fuel.—The number of heat units produced by the complete combustion of a unit weight of a fuel is called its *calorific value* or *heat of combustion*. In the case of a gaseous fuel the calorific value may also be given per unit volume of the gas at standard pressure and temperature. The calorific value of a fuel is determined by burning it in a fuel calorimeter in which the heat produced is communicated to a known weight of water. The amount of heat taken up by the water is its rise in temperature multiplied by its weight.

The calorific values of the principle combustible constituents of fuels are given in the table below. The values under the heading "at constant pressure" are the values to be used in dealing with the combustion in boiler furnaces because in such furnaces the pressure is constant and the gases are free to expand or contract. The reason for the difference between the calorific values at constant pressure and at constant volume and the calculation of this difference are discussed in the next Art.

Calorific Values of the Principal Constituents of Fuels.

Combustible.	At constant pressure.		At constant volume.	
	C.H.U.	B.Th.U.	C.H.U.	B.Th.U.
Hydrogen	34500	62100	34095	61371
Carbon burned to CO_2	8100	14580	8100	14580
" " CO	2416	4349	2439	4390
Carbon monoxide (CO)	2436	4385	2426	4367
Methane (CH_4)	13344	24019	13276	23897
Ethylene (C_2H_4)	12182	21928	12143	21857
Sulphur	2300	4140	2300	4140

Calorific values are subject to experimental errors and the values determined by different experimenters generally differ. The values given in the above table are based on those determined by Berthelot, Thomsen, and others.

79. Calorific Values at Constant Pressure and at Constant Volume.—When fuel is burned in ordinary furnaces it is burned at *constant pressure*, generally atmospheric, but in what are known as bomb calorimeters the fuel is burned in a confined space so that the volume of the products of combustion is the same as that of the fuel and oxygen before firing. The fuel is then said to be burned at *constant volume*. Now the same fuel may or may not have the same calorific value when burned under these different conditions.

Consider the combustion of hydrogen. The equation $2H_2+O_2=2H_2O$ shows that 2 volumes of hydrogen combine with 1 volume of oxygen to form 2 volumes of water vapour which in a calorimeter is condensed to water whose volume may be neglected. First imagine 2 volumes of hydrogen, denoted by $2v$, to be mixed with 1 volume of oxygen, denoted by v, and let there be an excess volume of oxygen denoted by x, the mixture to be in a long cylinder fitted with a piston. Let the pressure on the outer face of the piston be constant and equal to P the pressure of the mixture in the cylinder beneath the piston.

Now let the mixture of hydrogen and oxygen be ignited. An explosion will take place, the temperature will be greatly increased, and the piston being free to rise, the water vapour formed by the combustion of the hydrogen will increase in volume and the piston will rise until the volume beneath it is, say, V. External work equal to $P(V - 3v - x)$ will be done in pushing out the piston. Now let the heat be withdrawn by water surrounding the cylinder. The water vapour in the cylinder will condense and the piston will descend and in descending work will be done on the water vapour and excess oxygen until all the water vapour is condensed when the work done by the external pressure on the piston will be $P(V - x)$. The net external work done is therefore $- 3Pv$ and this is converted into heat which will be communicated to the water outside the cylinder. If the experiment be repeated but with the piston fixed in the cylinder so that it cannot rise or fall no external work will be done by or on the gases in the cylinder and the amount of heat evolved or communicated to the cooling water will therefore be less than in the first experiment by the heat equivalent of the work $3Pv$.

If P is in lb. per square foot and v is in cubic feet then $3Pv$ is in foot-pounds. The heat equivalent of the work $3Pv$ is $3APv$, where A is the reciprocal of the mechanical equivalent of heat. Taking P as standard atmospheric pressure $14{\cdot}7 \times 144$ lb. per square foot, and $2v$, the volume of hydrogen, used as $178{\cdot}6$ cubic feet per lb. at 0° C., and A as 1/1400. Then, $3APv = \dfrac{3 \times 14{\cdot}7 \times 144 \times 178{\cdot}6}{1400 \times 2} = 405$ C.H.U., and this is the difference between the values 34,500 and 34,095 given in the table on p. 105 for hydrogen at constant pressure and at constant volume respectively.

The student should in a similar manner verify the differences between the other values at constant pressure and at constant volume given in the table. Where there is no difference between the two calorific values the volume of the fuel and oxygen before burning is the same as that of the products of combustion at the same pressure after cooling to the same temperature.

When a calorific value is given without qualification as to constant pressure or constant volume, calorific value at constant pressure is to be understood.

80. Higher and Lower Calorific Values.—When hydrogen, or a fuel containing hydrogen, at ordinary atmospheric temperature, is burned and the products of combustion are cooled to the original temperature of the fuel, the water vapour formed by the combustion of the hydrogen is condensed and the whole of the heat of combustion is recovered or communicated to the cooling water, and the true calorific value of the fuel is determined either at constant pressure or at constant volume. When, however, a fuel is burned in ordinary furnaces the products of combustion escape at a temperature above that at which the water vapour would condense, and in addition to the sensible heat in the escaping gases there is a considerable amount of latent heat in the water vapour which is lost.

If the latent heat in the steam formed by the combustion of 1 lb. of hydrogen be deducted from the total heat produced by the combustion

of that hydrogen the result is called the *lower calorific value* of the hydrogen and the total heat of combustion is the *true* or *higher calorific value*. The latent heat referred to is the latent heat of dry saturated steam at the standard atmospheric pressure of 14·7 lb. per square inch.

The true heat of combustion of 1 lb. of hydrogen is 34,500 C.H.U. Now 1 lb. of hydrogen combines with 8 lb. of oxygen to form 9 lb. of steam and the latent heat of this steam is $9 \times 539 = 4851$ C.H.U. The lower calorific value of hydrogen is therefore $34{,}500 - 4851 = 29{,}649$ C.H.U.

This lower calorific value of hydrogen is very frequently taken in working out the results of boiler trials, but there are objections to this practice. The reason given for taking the lower calorific value is that the latent heat in the water vapour of the products of combustion is not available, but no more is a large amount of the sensible heat which escapes in the products of combustion. It is conceivable that a boiler plant might be designed in which the temperature of the products of combustion passing to the chimney might be reduced to atmospheric temperature, in which case all the heat both sensible and latent in the chimney gases due to the combustion of the fuel would be recovered.

On the other hand the efficiency of a boiler when using a fuel containing a large percentage of hydrogen will appear to be less than when using a fuel containing a small percentage of hydrogen and this lower efficiency is not the fault of the boiler. This will be made clear by considering two cases. In one case let the fuel used be pure carbon and in the other suppose that the fuel consists of 85 per cent. of carbon and 15 per cent. of hydrogen. In both cases suppose that the temperature of the waste gases is 300° C. and that the temperature of the fuel and air entering the furnace is 0° C. Also, in both cases let the fuel be burned with the minimum amount of air necessary for complete combustion.

In the first case the waste gases will consist of 3·67 lb. of CO_2 and 8·94 lb. of N per lb. of fuel, and the heat carried away by these gases will be

$$300(3{\cdot}67 \times 0{\cdot}22 + 8{\cdot}94 \times 0{\cdot}24) = 886 \text{ C.H.U.}$$

The calorific value of the fuel is 8100 C.H.U. per lb. and the heat available for use by the boiler is $8100 - 886 = 7214$ C.H.U. which is 89 per cent. of the heat of combustion of the fuel.

In the second case the waste gases will consist of 3·12 lb. of CO_2, 1·35 lb. of H_2O, and 11·61 lb. of N per lb. of fuel, and the heat carried away by these gases will be

$$300\,(3{\cdot}12 \times 0{\cdot}22 + 11{\cdot}61 \times 0{\cdot}24) + (639 + 0{\cdot}48 \times 200) \times 1{\cdot}35 = 2034 \text{ C.H.U.}$$

(Note that the heat carried away by 1 lb. of steam is made up of: sensible heat, 100; latent heat, 539; and superheat, $0{\cdot}48 \times 200$.)

The calorific value of the fuel is

$$0{\cdot}85 \times 8100 + 0{\cdot}15 \times 34{,}500 = 12{,}060 \text{ C.H.U. per lb.}$$

Heat available for use by the boiler $= 12{,}060 - 2034 = 10{,}026$ C.H.U. which is 83 per cent. of the true calorific value of the fuel.

If the latent heat of the steam, 539 × 1·35, be deducted from the heat in the waste gases and also from the calorific value of the fuel the heat available for use by the boiler will still be 10,026 C.H.U., but this is now 88 per cent. of the altered or lower calorific value of the fuel.

From the above it will be seen that if the boiler utilizes all the available heat in each case its efficiency is 89 per cent. with the carbon fuel, and only 83 per cent. with the carbon-hydrogen fuel using the higher calorific value of the fuel. But by using the lower calorific value the efficiency with the carbon-hydrogen fuel is raised to 88 per cent.

It will be observed that the greater calorific value of the fuel containing hydrogen is partly neutralized by the loss of heat in the greater weight of nitrogen in the waste gases and also by the specific heat of the steam being much higher than that of the other waste gases.

When the calorific value of hydrogen or of a fuel containing hydrogen is given without the qualification higher or lower the higher or true value is to be understood.

81. Calorific Values of Carbon.—It has been shown that 1 lb. of carbon combines with $2\frac{2}{3}$ lb. of oxygen to form $3\frac{2}{3}$ lb. of carbon dioxide (CO_2). The heat produced by this combination has been found by means of a calorimeter to be 8100 C.H.U. Again, 1 lb. of carbon combines with $1\frac{1}{3}$ lb. of oxygen to form $2\frac{1}{3}$ lb. of carbon monoxide (CO), but the heat produced by this combination cannot be determined in a calorimeter. Again, 1 lb. of carbon monoxide (CO) combines with $\frac{4}{7}$ lb. of oxygen to form $1\frac{4}{7}$ lb. of carbon dioxide (CO_2) and the heat produced by this combination has been found by burning the CO in a calorimeter, the result being 2436 C.H.U.

It will now be shown how the calorific value of carbon when burned to CO may be calculated from the above results. If 1 lb. of carbon be burned first to CO and the resulting CO be afterwards burned to CO_2 the total heat produced will be the same as if the 1 lb. of carbon had been burned directly to CO_2. Let x be the calorific value of 1 lb. of carbon when burned to CO. The weight of the CO from 1 lb. of carbon is $2\frac{1}{3}$ lb. Now the calorific value of 1 lb. of CO is 2436 C.H.U., therefore the heat produced by the combustion of $2\frac{1}{3}$ lb. of CO is $2\frac{1}{3} \times 2436 = 5684$ C.H.U. Hence,

$$x + 5684 = 8100 \qquad \text{and } x = 8100 - 5684 = 2416 \text{ C.H.U.}$$

82. Calorific Value of a Chemical Compound.—Generally, but not always, when two or more elements unite to form a chemical compound heat is evolved, and, conversely, to split up this compound into its constituent elements will require the expenditure of the same amount of heat.

Hence, it follows that in general the heat of combustion of a chemical compound will not be the same as that of the sum of the heats of combustion of the constituents of the compound. Consider the compound methane or marsh gas (CH_4) The molecular weight of CH_4 is $12 + 4 = 16$, and therefore 1 lb. of CH_4 contains $\frac{3}{4}$ lb. of C and $\frac{1}{4}$ lb. of H. Now $\frac{3}{4}$ lb. of C burning to CO_2 and $\frac{1}{4}$ lb. of H burning to H_2O will produce $\frac{3}{4} \times 8100 + \frac{1}{4} \times 34{,}500 = 14{,}700$ C.H.U.

But 1 lb. of CH_4 when burned to CO_2 and H_2O in a calorimeter produces 13,344 C.H.U., and the difference between these two results is 14,700 − 13,344 = 1356 C.H.U. This amount, 1356 C.H.U., is the amount of heat which must be expended in splitting up the compound CH_4 into carbon and hydrogen before the combustion of CH_4 to carbon dioxide and water vapour takes place.

83. Calorific Value Calculated from Analysis of Fuel.—Although the elements which enter into the composition of a fuel and the proportions in which they occur may be determined by chemical analysis, the way in which these elements are combined in the fuel is generally unknown; particularly is this true of coal and other solid fuels. Nevertheless an approximate value of the heat of combustion of a fuel may be found by calculation from its chemical analysis by assuming that the heat of combustion of the fuel is equal to the sum of the heats of combustion of its constituents.

When a fuel contains both hydrogen and oxygen it is a common practice, in calculating the calorific value of a fuel from its analysis, to assume that the oxygen is already in combination with part of the hydrogen in the proportion to form H_2O and that only the remaining part of the hydrogen, called the "free hydrogen," is available for the production of heat when the fuel is burned. But it must be understood that it is doubtful whether the oxygen is combined as assumed.

An example will serve to show how the calorific value of a fuel is calculated from its chemical analysis. Take the coal whose composition was given on p. 102 and is here repeated. C 76·6, H 5·3, O 12·8, N 2·1, S 1·2, and ash 2·0 per cent.

$$\text{“Free hydrogen” per lb. of coal} = 0{\cdot}053 - \frac{0{\cdot}128}{8} = 0{\cdot}037 \text{ lb.}$$

Heat produced by burning of free hydrogen	$= 0{\cdot}037 \times 34{,}500 =$	1276 C.H.U.
Heat produced by burning of carbon	$= 0{\cdot}766 \times 8100 =$	6205 „
Heat produced by burning of sulphur	$= 0{\cdot}012 \times 2300 =$	28 „
Total heat of combustion of 1 lb. of coal	$=$	7509 „
	$=$	13,516 B.Th.U.

If all the hydrogen in the above fuel is assumed to be available for the production of heat, the calorific value of the coal would be 8061 C.H.U. or 14,510 B.Th.U.

The most satisfactory way of finding the calorific value of any fuel is to burn a proper sample of it in a calorimeter.

84. Carbon Value and Evaporative Value of a Fuel.—The calorific value of a fuel divided by the calorific value of carbon when burned to CO_2 gives the *carbon equivalent* or *carbon value* of the fuel. For example: if the calorific value of a particular kind of coal is 7509 C.H.U. or 13,516 B.Th.U., then the carbon value of this coal is $\frac{7509}{8100}$ or $\frac{13516}{14580} = 0{\cdot}927$. This means that 0·927 lb. of carbon will produce the

same amount of heat as 1 lb. of the coal when combustion is complete.

A very convenient measure of the heating value of a fuel is the weight of water which 1 lb. of the fuel would evaporate if all the heat produced by the combustion of the fuel were given to the water. To make this measure quite definite it is obvious that the temperature of the water and the temperature of the steam must be standard temperatures. Both these temperatures are taken as that of boiling water at the standard atmospheric pressure (14·7 lb. per square inch), that is 100° C. or 212° F. 1 lb. of water at 100° C. or 212° F. evaporated at that temperature to dry saturated steam is the standard evaporation unit and represents 539 C.H.U. or 970 B.Th.U. The *evaporative value* or *evaporative power* of a fuel is therefore its calorific value in C.H.U. divided by 539, or its calorific value in B.Th.U. divided by 970. For example: the evaporative value of a coal whose calorific value is 7509 C.HU. or 13,516 B.Th.U. is $\frac{7509}{539}$ or $\frac{13516}{970} = 13{\cdot}93$ lb.

85. Mean Specific Heat of Products of Combustion.—If w_1, w_2, w_3, etc., be the relative weights of the individual products of the combustion of a fuel and if s_1, s_2, s_3, etc., be the specific heats of these individual products respectively then the mean specific heat of the products of combustion is

$$s = \frac{s_1w_1 + s_2w_2 + s_3w_3 + \text{etc.}}{w_1 + w_2 + w_3 + \text{etc.}}$$

The various specific heats s_1, s_2, s_3, etc. vary with the temperature, being greater the higher the temperature. For temperatures up to that of the chimney gases from a steam boiler the following values of the specific heats may be used:

Air, nitrogen, and carbon monoxide (CO) . . .	0·24
Oxygen, and carbon dioxide (CO_2)	0·22
Water vapour above 100° C.	0·48
Ash	0·20

At the temperatures occurring in the furnace itself the values of the specific heats are not known with certainty.

86. Temperature of Combustion.—Assuming that when a fuel is burned the whole of the heat of combustion is, in the first instance, given to the products of combustion, and assuming values of the mean specific heats of the individual gases in the products, the *temperature of combustion* or *calorific intensity* is readily calculated as shown by the following examples.

A fuel at 0° C. containing 85 per cent. of carbon and 15 per cent. of hydrogen, by weight, is completely burned. It is required to calculate the temperature of combustion: (*a*) when the air supplied is the minimum necessary for complete combustion; (*b*) when the air supplied is double that necessary for complete combustion.

Consider the combustion of 1 lb. of the fuel, and let t = temperature of combustion in degrees C.

(*a*) 0·85 lb. of carbon requires 2·27 lb. of oxygen to form 3·12 lb. of CO_2.

0·15 lb. of hydrogen requires 1·20 lb. of oxygen to form 1·35 lb. of H_2O.

Total oxygen required by 1 lb. of fuel = 3·47 lb.

Weight of nitrogen associated with this oxygen $= 3{\cdot}47 \times \frac{77}{23} = 11{\cdot}61$ lb.

Heat of combustion $= 0{\cdot}85 \times 8100 + 0{\cdot}15 \times 34{,}500 = 12{,}060$ C.H.U.

Assume that the mean specific heats of N and CO_2 are 0·27 and 0·28 respectively from 0° to t°. Take the mean specific heat of steam at atmospheric pressure as 0·6 from 100° to t°.

$$\begin{aligned} \text{Heat in}\quad N &= 11{\cdot}61 \times 0{\cdot}27t = 3{\cdot}135t. \\ ,,\quad CO_2 &= 3{\cdot}12 \times 0{\cdot}28t = 0{\cdot}874t. \\ ,,\quad H_2O &= 1{\cdot}35(100 + 539) + 1{\cdot}35 \times 0{\cdot}6(t - 100) \\ &= 782 + 0{\cdot}810t. \end{aligned}$$

Total heat in products $= 782 + 4{\cdot}819t = 12{,}060$.

Therefore $t = 2340°$ C. = 4244° F.

(*b*) With 100 per cent. of excess air the resulting temperature will be lower and the mean specific heats of N, CO_2 and O may be taken as 0·26, 0·27, and 0·23 respectively from 0° to t°. The mean specific heat of the steam may be taken as 0·5 from 100° to t°.

The products will consist of 23·22 lb. of N, 3·47 lb. of O, 3·12 lb. of CO_2, and 1·35 lb. of H_2O.

$$\begin{aligned} \text{Heat in}\quad N &= 23{\cdot}22 \times 0{\cdot}26t = 6{\cdot}037t. \\ ,,\quad O &= 3{\cdot}47 \times 0{\cdot}23t = 0{\cdot}798t. \\ ,,\quad CO_2 &= 3{\cdot}12 \times 0{\cdot}27t = 0{\cdot}842t. \\ ,,\quad H_2O &= 1{\cdot}35(100 + 539) + 1{\cdot}35 \times 0{\cdot}5(t - 100) \\ &= 795 + 0{\cdot}675t. \end{aligned}$$

Total heat in products $= 795 + 8{\cdot}352t = 12{,}060$.

Therefore $t = 1349°$ C. = 2460° F.

Comparing cases (*a*) and (*b*) it is seen that the excess air has an important influence in reducing the temperature of combustion. The actual temperatures in practice are generally much lower than those calculated as above because of the loss of heat by radiation and also because of partial dissociation of the products at high temperatures. The temperature of combustion is also considerably influenced by the rate at which the fuel is burned, being lower the lower the rate of combustion. In ordinary boiler furnaces the actual temperature varies from about 800° C. (1472° F.) to a maximum of about 1650° C. (3002° F.).

87. Varieties of Fuel.—Fuels may be classified as: (1) *Solid fuels*, such as, wood, peat, and coal; (2) *Liquid fuels*, such as, petroleum, shale oil, tar, and alcohol; (3) *Gaseous fuels*, such as, natural gas, blast furnace gas, coal gas, and producer gas.

88. Wood.—Thoroughly dried wood contains about 50 per cent. of carbon, 6 per cent. of hydrogen, 42 per cent. of oxygen, and 2 per cent. of ash. On account of the large amount of oxygen present less than 1 per cent of the hydrogen is available for combustion, and the heat of combustion of wood is almost entirely due to the carbon which it contains.

Ordinary air-dried firewood usually contains about 20 per cent. of moisture, so that its composition per cent. is about as follows: carbon, 40; hydrogen, 4·8; oxygen, 33·6; ash, 1·6; and water, 20. Newly felled wood contains from 25 to 50 per cent. of moisture.

The average calorific value of air-dried wood with 20 per cent. of moisture is about 3300 C.H.U. (5940 B.Th.U.)

89. Peat.—Peat consists of the fossil remains of vegetable matter, generally mosses and aquatic plants. Ordinary air-dried peat usually contains from 20 to 30 per cent. of water.

The mean composition of perfectly dry peat is about as follows: carbon, 58; hydrogen, 6; oxygen, 30; nitrogen, 1; and ash, 5.

The composition per cent. of air-dried peat with 25 per cent. of moisture is about as follows: carbon, 43·5; hydrogen, 4·5; oxygen, 22·5; nitrogen, 0·8; ash, 3·8; and water, 25. The calorific value of this peat is about 3500 C.H.U. (6300 B.Th.U.).

90. Coal.—Coal is the product of vegetable matter which has, during the course of ages, been decomposed and solidified under great pressure. The character of the coal depends on the length of time which has been occupied in its production and on the amount of pressure and heat to which it has been subjected in the strata of the earth.

The principal varieties of coal are as follows:—I. Lignite or brown coal. II. Bituminous coals, which embrace (1) cannel or long flaming coal, (2) caking coal, and (3) non-caking or dry bituminous coal. III. Anthracite.

The gradual conversion of woody fibre into peat and the different kinds of coal is shown in the following table, given on the authority of Dr. Percy:—

	Carbon.	Hydrogen.	Oxygen.
Wood	100	12·18	38·07
Peat	100	9·85	55·67
Lignite	100	8·37	42·42
Bituminous coal . . .	100	6·12	21·23
Anthracite (Wales) . .	100	4·75	5·28
Anthracite (Pennsylvania)	100	2·84	1·74

Lignite or brown coal is intermediate in appearance and properties between peat and true coal. It burns with a very long smoky flame, and it is generally non-caking. After drying in the air lignite contains from 15 to 20 per cent. of moisture. If thoroughly dried in a stove and again exposed to the air it reabsorbs the water which it lost in drying. The composition of lignite varies considerably. A sample of good quality when thoroughly dried contained 65 per cent. of carbon, 5 per cent of hydrogen, 22 per cent. of oxygen, 1 per cent. of nitrogen, and 7 per cent. of ash. The specific gravity varies from 1·2 to 1·3.

Cannel coal is highly valued for gas-making. It burns with a long flame, and gives off large quantities of smoke. Specific gravity 1·27 to 1·32.

Caking bituminous coal softens and swells when heated, and the parts adhere together, forming a pasty mass. It burns with a fairly long flame, and requires careful stoking to avoid smoke. Specific gravity 1·26 to 1·36.

Non-caking or dry bituminous coal burns with a shorter flame than that of the caking coal, and it gives off little or no smoke. Specific gravity 1·28 to 1·42.

Anthracite burns without flame or smoke, and with an intense local heat, but it requires a strong draught for its combustion. It is hard and brittle, and most varieties decrepitate considerably when heated, especially when the heat is applied suddenly. Care has therefore to be taken that the fire is so managed that the small pieces do not fall through between the fire-bars and get lost. Specific gravity 1·35 to 1·7.

The moisture in coal, other than brown coal, when brought to the surface may be from 5 to 10 per cent. After exposure to the air the amount of moisture may be from 1 to 5 per cent.

The composition, per cent., of some typical coals is given in the following table, but it must be remembered that the composition of coal of the same class and even from the same pit is variable.

Class and Locality of Coal.	C	H	O	N	S	Ash.
Cannel (Wigan)	84·1	5·7	7·8	—	—	2·4
Cannel (Scotland)	66·5	7·5	10·8	1·4	0·8	13·0
Caking Bituminous (Northumberland)	81·4	5·8	7·9	2·1	0·7	2·1
Caking Bituminous (South Wales)	83·4	5·7	5·9	1·7	0·8	2·5
Non-caking Bituminous (South Wales)	87·6	4·4	2·5	1·1	1·1	3·3
"Nixon's Navigation" (Glamorganshire)	88·8	4·1	2·4	1·0	0·7	3·0
Anthracite (South Wales)	90·4	3·3	3·0	0·8	0·9	1·6
Anthracite (Pennsylvania)	92·6	2·6	1·6	0·9	—	2·3

91. Artificial Solid Fuels.—*Briquette fuel* is usually made by mixing coal dust with pitch or some other binding material, the mixture being pressed and formed into hard blocks of rectangular shape. Good briquette fuel contains about 7 per cent. of ash, 8 per cent. of pitch, and 3 per cent. of moisture in addition to the coal. Its calorific value is about 8000 C.H.U. (14,400 B.Th.U.).

Briquette fuel is also made from peat and from brown coal and has then a calorific value of 4000 to 5000 C.H.U. (7200 to 9000 B.Th.U.).

Wood charcoal is made by heating wood out of contact with the atmosphere, or with only a limited supply of air, to a temperature not lower than 550° Fahr. The higher the temperature, the blacker and harder is the charcoal produced. The yield of charcoal varies from 15 to 25 per cent. by weight of the wood from which it is produced, the yield being lower the higher the temperature. Dry charcoal contains from 80 to 95 per cent. of carbon, 0·5 to 3 per cent. of available hydrogen, and 1 to 5 per cent. of ash, the remainder being nitrogen and combined oxygen and hydrogen. Charcoal which has been exposed to the air usually contains from 5 to 12 per cent. of moisture. The calorific value of good dry charcoal is about 7000 C.H.U. (12,600 B.Th.U.).

Peat charcoal is prepared from peat in the same manner that wood charcoal is made from wood. Good peat charcoal when perfectly dry contains from 80 to 90 per cent. of carbon, and 10 to 15 per cent. of ash. It is usually extremely friable.

Coke is the solid carbonaceous material left after coal has been heated to a high temperature with a limited supply of air, or, in the case of gas coke, with no air at all.

The best coke for fuel is prepared from bituminous coals. It is hard, brittle, and porous, of a dark grey colour and slightly metallic lustre.

The yield of coke from bituminous coals is from 50 to 80 per cent. of the weight of the coal. Anthracite yields from 80 to 95 per cent. of coke, which is of a pulverulent or powdery nature and is not valued as a fuel.

Good dry coke contains from 85 to 95 per cent. of carbon, 0·25 to 2 per cent. of sulphur, and 4 to 12 per cent. of ash. Exposed to the air, it absorbs from 10 to 20 per cent. of moisture.

The calorific value of coke varies from 6600 to 7600 C.H.U. (11,880 to 13,680 B.Th.U.).

92. **Liquid Fuel.**—For steam raising the liquid fuel almost exclusively used is obtained from the natural mineral oil *petroleum*. The chief sources of the supply of petroleum are the oil wells of the United States of America and Russia, but smaller quantities are also obtained from Mexico, Borneo, Rumania, and other countries.

The crude petroleum as it comes from the well usually contains from 83 to 87 per cent. of carbon and from 11 to 14 per cent. of hydrogen together with small percentages of oxygen, nitrogen, and sulphur. Its specific gravity varies in general from 0·8 to 0·95, and its lower calorific value is about 10,800 C.H.U. (19,440 B.Th.U.).

The crude petroleum is a mixture of many hydrocarbons having different densities and different boiling points.

The liquid fuel generally used for steam raising is the residue obtained when the crude oil is subjected to partial distillation at temperatures up to 300° C. (572° F.). During this partial distillation the lighter constituents evaporate at the lower temperatures. Petrol or gasolene comes off at temperatures from 70° C. to 80° C. The different grades of naphtha come off at temperatures from 80° C. to 150° C. Paraffin oil or kerosene comes off at temperatures from 150° C. to 300° C. At the different temperatures the vapours which come off are liquefied as they pass through water-cooled pipe coils. The residue or fuel oil has a specific gravity of 0·88 to 0·94, and a calorific value of 10,100 to 10,900 C.H.U. (18,180 to 19,620 B.Th.U.). Its flash point is about 150° C. (302° F.).

The *flash point* of an oil is the temperature at which it begins to give off inflammable vapour. The British Admiralty specification for Navy oil fuel limits the flash point to 175° F. (about 80° C.). In the British Mercantile Marine the minimum flash point is 150° F. (about 66° C.).

In the following table the characteristics of a good steam coal and fuel oil are summarized for comparison.

			Coal	Oil
Composition	Carbon	per cent.	88·0	86·5
	Hydrogen	„	4·3	11·5
	Oxygen	„	2·4	1·0
	N., S., and Ash	„	5·3	1·0
Minimum weight of air for 1 lb. of fuel		lb.	11·5	14·0
Calorific value (higher)		C H.U.	8,508	10,931
		B.Th.U.	15,314	19,676
Theoretical evaporative power in lb. of water from and at 100° C. per lb. of fuel			15·8	20·3
Approximate space occupied by 1 ton of fuel		c. ft.	44	39

93. Burners for Liquid Fuel.—The functions of a burner for liquid fuel are to direct the fuel into the furnace and to atomize it, that is, to break it up into a fine spray. The atomizing is generally done in one of three ways: (*a*) by steam jet, (*b*) by air jet, (*c*) by delivering the oil under high pressure through a specially designed orifice, the oil being led to the orifice through a passage or passages of special form. As examples of burners operating in these three different ways Kermode's liquid fuel burners will now be illustrated and described.

The *steam-jet burner* is shown in longitudinal section in Fig. 83. The oil enters at A and has a whirling motion given to it by the long spiral B formed on a prolongation of the spindle C of the valve V.

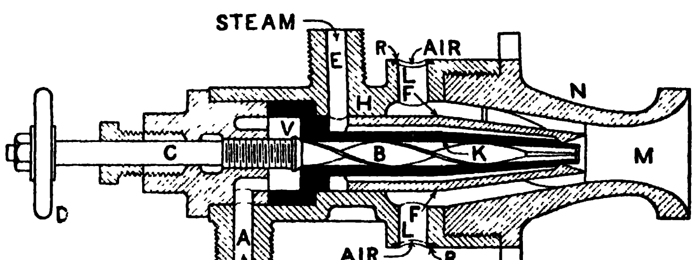


FIG. 83.—Kermode's steam-jet liquid fuel burner.

The supply of oil is regulated by the valve V which is operated by the hand wheel D. The steam enters at E and, passing through slots in the cylindrical part of the hollow cone F where it fits into the casing H, it travels through the annular space between the hollow cone F and the interior hollow cone K which surrounds the spiral prolongation of the valve spindle. It will be seen that the whole of the oil as it passes through the burner is steam-jacketed. At the front end of the burner there is a nozzle N which is screwed into the casing H, and between this nozzle and the hollow cone F there is an annular passage in which are spiral guides. By the inductive action of the steam, the air, which enters through holes L in the casing, is drawn through this annular passage. The whirling oil together with the steam and the whirling air mix at M where the oil is atomized and the mixture being ignited burns with a long flame which projects into the boiler furnace.

The opening between the front ends of the hollow cones F and K is regulated by rotating the nozzle N which carries with it the hollow cone F.

The supply of air is regulated by means of the movable perforated ring R which acts exactly as in the ordinary Bunsen burner.

Since the supplies of oil, steam, and air may be regulated independently the same burner may be used for different powers within wide limits.

The amount of steam used by steam-jet burners is about $3\frac{1}{2}$ per cent. of the total steam raised by the liquid fuel passing through them, and the heat utilized by the boiler is from 70 to 75 per cent. of the heat of the fuel.

An objection to the use of steam for atomizing the liquid fuel on board ship is that the steam so used cannot be condensed and returned to the boiler, and this means a considerable addition to the feed water which has to be made up by the evaporation and condensation of sea water. The steam used in the burner also increases the weight of the waste gases and therefore adds to the heat lost in these gases.

The *air-jet burner* is shown in longitudinal section in Fig. 84. The oil enters at A and its supply to the burner is controlled by the valve

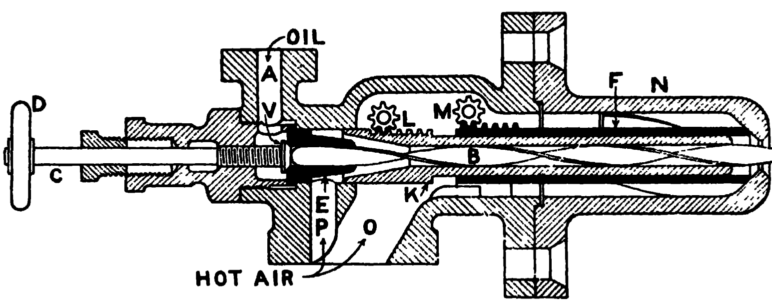


FIG. 84.—Kermode's air-jet liquid fuel burner.

V operated by the hand wheel D on the end of the valve spindle C. The air to be used in the burner is previously heated by passing through a heater placed in the hot gases from the boiler. The heated air enters the burner by the passages O and P. The portion of the air which enters by the passage P mixes with the oil after it has passed through the valve V. This oil and air travel on together, the oil being rapidly vaporized by the hot air mixing with it. The mixing of the air with the oil and the vaporization of the latter are assisted by giving them a whirling motion by means of the long spiral B which is formed on a prolongation of the valve spindle and contained by the tube K. The tube K is surrounded by another tube F, and these tubes are provided with racks into which gear the pinions L and M as shown.

The orifice R through which the hot air and vaporized oil finally pass on leaving the burner is formed in the end of an extension N of the casing. The hot air which enters by the passage O travels to the orifice R through the annular space between the casing and the tube F.

This annular passage is provided with spiral guides which give a whirling motion to the air passing over them. The opening over the front end of the tube F, through which the air entering at O has to pass before escaping through the orifice R, is regulated by the pinion M. Also the opening between the back end of the tube K and the oil-delivering nozzle E is regulated by the pinion L and the air entering at P is thus controlled. Combustion commences at the orifice R.

The air entering the passages O and P is supplied by an air compressor at a pressure which generally does not exceed 4 lb. per square inch. This air is of course heated after it leaves the compressor. In addition to the compressed air supply there is a further supply of air at atmospheric pressure, outside the burner, induced by the draught in the furnace.

Less than 2 per cent. of the steam raised is required to drive the air compressor and if this is afterwards condensed no fresh feed water is lost. It is claimed that with this type of burner 84 per cent. of the heat of the fuel is utilized in raising steam.

The *pressure-jet burner* is shown in longitudinal section in Fig. 85. H is the casing, or outer barrel, to a branch of which is coupled the

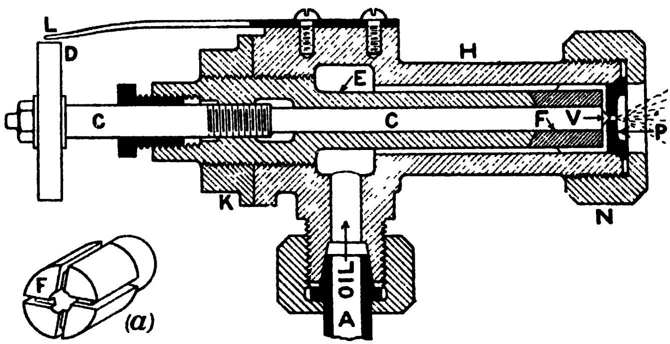


FIG. 85.—Kermode's pressure-jet burner.

oil pipe A. A tubular piece or inner barrel E is screwed into the back end of the casing H and locked by the nut K. A detachable piece F, called a spool, forms a continuation of the piece E. A pictorial view of the spool F is shown at (*a*). For a part of its length F is enlarged to fit the interior of the outer barrel H and its front end abuts against an orifice plate P which is secured by the cap-nut N as shown. In the enlarged part of F there are longitudinal grooves and its front end is also grooved. The grooves on the end of F are not radial but are tangential to a central circle on it. C is a central spindle screwed into the back end of E. The part of C in front of the screw fits the interior of E. On the front end of C is formed a conical valve V whose seat is on the orifice in the plate P. The opening of the valve V is regulated by the hand wheel D, the extent of the opening being shown by a fixed pointer L and graduations on the circumference of D.

The oil fuel is pumped to the burner under a pressure which is generally about 150 lb. per square inch. Before entering the burner

the oil fuel is heated to a temperature of about 95° C. (203° F.) but the temperature to which it is heated depends upon the character of the fuel used. An oil fuel of an asphalte base requires more heating than one with a petroleum base.

Entering the burner at A the oil passes along the annular passage between E and H and through the grooves on F to the orifice in P. Each burner is provided with a number of spools and orifice plates. The grooves on the different spools and the orifice in the corresponding orifice plate are made to suit different rates of combustion.

Fig. 86 shows how the burner is fitted to a boiler furnace and how the air for the combustion of the oil is admitted and regulated. A cylindrical casing attached to the furnace front is provided with rectangular openings fitted with shutters D. These shutters are hinged eccentrically in such a way that the air flowing into the furnace opens them, but any pressure from the inside due, say, to the bursting of a boiler tube, automatically closes them. The angular opening of the shutters is limited and regulated by adjustable stops E.

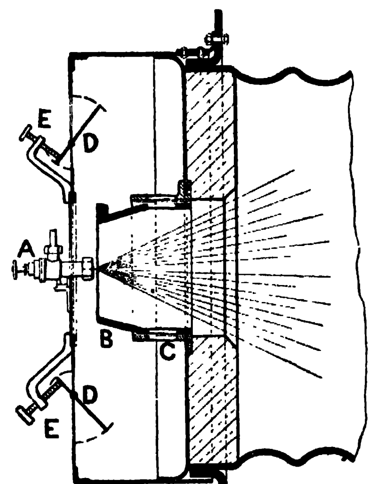


FIG. 86.

A is the burner or atomizer which discharges the oil spray through the tube B into the furnace. Air for the combustion of the fuel enters partly through ports at C which may be placed more or less opposite to similar ports in another tube surrounding B. These ports are inclined to the radial direction so that the air passing through them has a whirling motion given to it, while the air entering at the mouth of B travels in a direction parallel to the axis of the burner. The position of the tube B in relation to the outer tube may be adjusted from the stokehold while the burner is in operation.

With the pressure jet system it is claimed that 83 per cent. of the heat value of the fuel used can be utilized in actual work. The steam for operating the oil pump and for heating the oil fuel is condensed so that there is no loss of feed water. The steam used for these operations amounts to less than one per cent. of the steam raised.

Exercises V

1. Carbon is burned to carbon dioxide in a furnace, and the amount of air supplied is just sufficient for this purpose. Determine: (*a*) The percentage composition of the chimney gases, by weight. (*b*) The percentage composition of the chimney gases, by volume. (*c*) The volume of the gases, in cubic feet, passing up the chimney per pound of carbon burned if their temperature is 280° C. and their pressure 14·6 lb. per square inch. Volume of 1 lb. of hydrogen at 0° C. and 14·7 lb. per square inch = 178·6 cubic feet.

2. Same as exercise 1 except that the air supply is 50 per cent. greater than is necessary for the complete combustion of the carbon.

3. A fuel oil contains 85 per cent. of carbon and 15 per cent. of hydrogen by weight. Determine the minimum weight of oxygen and the minimum weight of air required for the complete combustion of 1 lb. of this oil. If the air actually supplied is 20 per cent. in excess of the minimum required, find the percentage composition of the products of combustion by volume, neglecting the volume of the condensed water vapour.

4. The percentage composition of a sample of anthracite was found by analysis to be: C 90·0, H 3·3, O 3·0, N 0·8, S 0·9, and ash 2·0. Calculate the minimum weight of air for the complete combustion of 1 lb. of this fuel. If 50 per cent of excess air is supplied find the percentage composition of the dry flue gases by volume.

5. In the combustion of, say, CH_4, state exactly the various meanings of the equation $CH_4 + 2O_2 = CO_2 + 2H_2O$.

A cubic foot of coal gas contains 0·34 cubic foot of C_2H_4. How much oxygen and how much air are needed for the complete combustion of this portion of the coal gas? [B.E.]

6. A gas used in an internal combustion engine had the following composition by *volume*: Hydrogen, 45 per cent.; Marsh gas (CH_4), 36 per cent.; Carbon monoxide, 15 per cent.; Nitrogen, 4 per cent. Find the volume of air required for the combustion of 1 cubic foot of the gas. (Oxygen in air is 21 per cent. by volume.) [Inst. C.E.]

7. The volume analysis of a producer gas is: H 14 %; CH_4 2 %; CO 22 %; CO_2 5 %; O 2 %; N 55 %. Find the air required for the perfect combustion of 1 cubic foot of the gas. If 40 % excess air is supplied, find the volume analysis of the dry products. Air contains O 20·9 %, N 79·1 % by volume. [U.L.]

8. In a boiler trial the dry coal as burned contained 84 per cent. of carbon and 3 per cent. of free hydrogen. The flue gas analysis gave 11·5 per cent. of carbon dioxide, 8·1 per cent. of oxygen, and 80·1 per cent. of nitrogen. Calculate per pound of dry fuel, the weight of necessary air and the weight of excess air. [U.L.]

9. In a boiler trial the following analysis of the flue gases was obtained: by volume, carbon dioxide 13 per cent., oxygen 6 per cent. Find the pounds of air per pound of coal if the carbon in the coal was 85 per cent. (Air contains 23 per cent of oxygen by weight.) [Inst. C.E.]

10. The analysis of the coal in a boiler trial was: C 88 %, H 3·6 %, O 4·8 %, other matters 3·6 %, and the flue gas analysis by volume was, CO_2 10·9 %, CO 1 %, O 7·1 %, N 81 %. Find the proportion of carbon burned to CO, and the air required per pound of fuel for the combustion as it actually occurred, and also the weight of flue gases per pound of fuel burned. [U.L.]

11. In a boiler trial the composition of the dry waste gases (by weight) was found to be: N 75·5 %, CO 0·5 %, CO_2 11·5 %, and O 12·5 %. The composition of the coal (by weight) was: C 88·4 %, H 3·6 %, O 2·6 %, N 0·6 %, S 0·8 %, ash 3·2 %, and water 0·8 %. Find the weight of air passing into the furnace per lb. of coal and also the minimum weight of air for complete combustion.

12. Estimate the higher calorific value and evaporative power of 1 lb. of petrol whose composition is: carbon 85 per cent., hydrogen 15 per cent., by weight. What is the minimum weight of air necessary for the complete combustion of 1 lb. of this petrol? If the excess air is 40 per cent. of the minimum required for complete combustion, what is the percentage composition of the products of combustion, by weight? Also, what is the percentage composition of the *dry* products, by volume?

13. Same as exercise 12 for 1 lb. of alcohol whose composition is: Carbon 52·2 per cent., hydrogen 13 per cent., and oxygen 34·8 per cent., by weight.

14. Calculate the higher and lower calorific values and evaporative powers of a coal whose percentage composition is: Carbon 80, hydrogen 5, oxygen 8, nitrogen 1·5, sulphur 1·1, and ash 4·4.

15. A producer gas has the following percentage analysis by volume: Hydrogen, 16; carbon monoxide, 20; carbon dioxide, 6; nitrogen, 58. Determine (*a*) its calorific value per c. ft. at standard temperature and pressure; (*b*) the minimum amount of air for complete combustion; (*c*) the volumetric analysis of the products if combustion is complete. Calorific value of 1 lb. of carbon burning to CO_2, 14,500; burning to CO, 4400; of hydrogen, 62,000 B.Th.U.

Composition of air by volume: Oxygen, 21 per cent.; nitrogen, 79 per cent. Volume occupied by 2 lb. of hydrogen is 357 c. ft. at standard temperature and pressure. [U.L.]

16. The *higher* or *true* calorific value of olefiant gas (C_2H_4) determined by a calorimeter is 12,182 C.H.U. per lb. Calculate the *lower* calorific value of this gas.

17. Carbon dioxide (CO_2), produced in the lower part of a furnace, in passing over red-hot carbon higher up is converted into carbon monoxide (CO) as shown by the equation $CO_2 + C = 2CO$. Calculate the chilling action in the part of the furnace where the reaction takes place as measured by the loss of heat per lb. of CO_2.

18. Calculate the theoretical maximum temperature of combustion of a coal at 0° C. whose percentage composition is: C, 87·0; H, 4·5; O, 5·5; ash, 3·0. Assume 50 per cent. excess air and that the mean specific heats of the products are:—N, 0·27; O, 0·24; CO_2, 0·28; ash, 0·23 from 0° C. to t° C.; and steam, 0·6 from 100° C. to t° C., where t is the temperature of combustion.

19. In a marine boiler of the Scotch type, coal having a calorific value of 8400 C.H.U. (15,120 B.Th.U.) per lb. is used. The funnel gases weigh 20 lb. per lb. of coal. The temperature of the air in the stokehold is 25° C. (77° F.) and the temperature of the gases in the uptake is 360° C. (680° F).

Calculate the temperature of the gases at the furnace bridge and at the entrance to the tubes on the following assumptions. Mean specific heats of gases in uptake, at entrance to tubes, and at furnace bridge, 0·25, 0·26, and 0·27 respectively. Four-fifths of the combustion takes place in the furnace and the remaining one-fifth in the combustion chamber. Of all the heat given to the boiler, one-third is delivered in the furnace, one-third in the combustion chamber, and one-third in the tubes and smoke-box.

20. The calorific value of a pound of oil of the kind used to fire a steam boiler is 10,000 C.H.U., and its density is 58 lb. per cubic foot. The calorific value of a pound of coal is 8000 C.H.U., and its density is 52 lb. per cubic foot. Express in horse-power-hours the energy which can be stored in a ship with 40,000 cubic feet bunker or tank capacity; first, if coal only is carried; secondly, if oil only is carried. [B.E.]

CHAPTER VI

STEAM BOILERS

94. Classification of Boilers.—Steam boilers may in general be divided into two main classes. (1) Boilers in which there is an external shell, generally in whole or in part cylindrical, which contains a fire-box or one or more tubes large enough to hold a grate upon which the fuel is burned. From the fire-box or furnace tube the products of combustion are led through the shell by one or more large tubes or by a considerable number of small tubes to a smoke-box or to flues outside the shell through which the gases pass to the chimney. Boilers of this class are sometimes called *tank boilers*. In boilers of this class which contain tubes the water is generally outside the tubes while the hot gases are inside, and from this condition these boilers are called *smoke-tube boilers*. (2) Boilers which contain or consist mainly of a large number of comparatively small tubes through which the water circulates, the fire and hot gases being on the outside of those tubes. Boilers of this class are called *water-tube boilers*.

The principal types belonging to the tank or smoke-tube class are, ordinary vertical boilers, Cornish and Lancashire boilers, Scotch marine boilers, and boilers of the locomotive type.

As to the relative numbers of the different types of steam boilers in use Mr. Edward G. Hiller, chief engineer of the National Boiler and General Insurance Co., gave some interesting figures in a paper read before the Manchester Association of Engineers in January, 1914. Mr. Hiller estimated that in 1913 the proportions of different types of boilers in use on land in the United Kingdom were as follows:

Lancashire boilers	32·6	per	cent.
Cornish boilers	12·7	,,	,,
Vertical boilers [1]	24·3	,,	,,
Locomotive type boilers[2]	18.6	,,	,,
Water-tube boilers	5·3	,,	,,
Other types of boiler	6·5	,,	,,

The above figures were obtained from a classification of the boilers insured with the National Boiler and General Insurance Co.

[1] Including vertical multitubular boilers as well as the ordinary vertical boiler with internal fire-box with cross tubes.

[2] Not including locomotives on the main lines of railways but including works shunting locomotives, traction engines, road rollers, steam motor wagons, portable locomotives on farms, and fixed locomotive-type boilers.

As Mr. Hiller pointed out in his paper, the figures for the water-tube boiler probably do not show sufficiently the increasing importance of the water-tube boiler as compared with the Lancashire boiler because many of the newer water-tube boilers which count only as one boiler have rated evaporations equal to that of several Lancashire boilers.

95. Simple Vertical Boilers.—In its simplest form the vertical boiler consists of a cylindrical shell AA (Fig. 87) surrounding a nearly cylindrical fire-box BB in the bottom of which is the grate. A tube C called the uptake passes from the crown of the fire-box to the crown of the shell and on the top of this uptake is placed the chimney. To increase the heating surface, and improve the circulation of the water, the fire-box is fitted with one or more cross-tubes FF. These cross-tubes are either flanged and riveted to the fire-box as in Fig. 87 or they are welded to it. The cross-tubes are placed slightly inclined to ensure a more efficient cir-

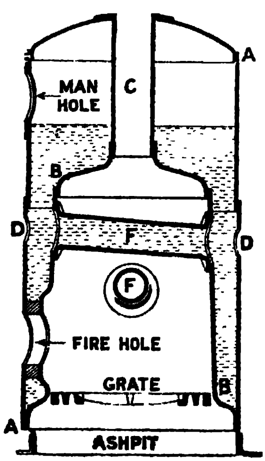


Fig. 87.

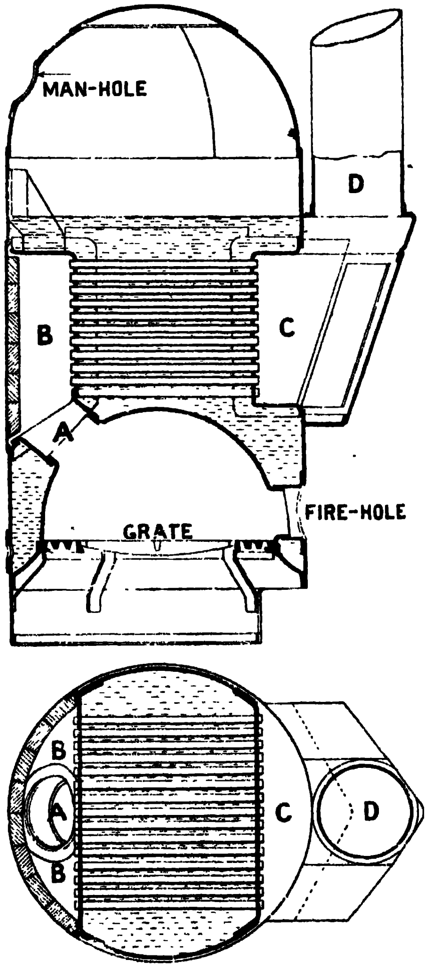


Fig. 88.

culation of the water. There are handholes D in the shell opposite to the ends of each cross-tube to give access for cleaning these tubes. There should also be several handholes in the shell at the bottom of the water space surrounding the fire-box.

A boiler of the type shown in Fig. 87 having a shell 3 feet 9 inches

in diameter and 9 feet high, and having three 9-inch cross-tubes in the fire-box, would have a grate area of 8·4 square feet and a heating surface of about 80 square feet.

96. Vertical Multitubular Boilers.—There are numerous designs of vertical multitubular boilers in use. One of the best known designs is the Cochran boiler shown in Fig. 88. It will be observed that the crowns of the fire-box and external shell are of hemispherical shape which, for a given weight of material in the form of plates, gives the maximum volume of space enclosed and also maximum strength.

The hot gases from the fuel pass from the fire-box through the short flue-pipe A into the combustion chamber B and from thence through numerous horizontal tubes to the smoke-box C from which they pass into the chimney D. The combustion chamber B is lined with fire-brick on the side next the shell.

A Cochran boiler having a shell 6 feet 6 inches in diameter and 14 feet 6 inches high would have a grate area of about 22 square feet and a total heating surface of about 500 square feet. This boiler would contain 165 tubes $2\frac{1}{2}$ inches outside diameter presenting about 428 square feet of heating surface, and, constructed for a working pressure of 100 lbs per square inch, it would weigh about $7\frac{3}{4}$ tons when complete with mountings, but without water.

Vertical boilers have the advantage of taking up a comparatively small amount of floor area.

97. Lancashire and Cornish Boilers.—The Lancashire boiler has a cylindrical shell usually from 6 feet to 9 feet in diameter and from 24 feet to 30 feet long. Modern Lancashire boilers are commonly 8 feet in diameter and 30 feet long. The shell is traversed internally by two flue tubes containing, at their front ends, the furnaces. This type of boiler is set in brickwork forming external flues so that part of the heating surface is on the external shell.

The Cornish boiler differs from the Lancashire in having only one internal flue tube instead of two and in being generally of smaller dimensions. The shell of the Cornish boiler is usually from 4 feet to 6 feet in diameter and its length from 14 feet to 24 feet. The Cornish boiler has a brickwork setting similar to that of the Lancashire boiler.

The main features of the Lancashire boiler and its brickwork setting are shown in Figs. 89 and 90. AB is the external shell traversed by the two internal flue tubes CD. The main parts of these flues are made as large in diameter as possible in order to get in grates of sufficient area and of minimum length, but at the back ends they are reduced in diameter to provide access to the lower part of the boiler and also to allow room for the joint between the back end plate and the cylindrical shell being placed inside, the corresponding joint at the front end being on the outside of the shell.

At the back of the grate is the fire bridge E. The hot gases pass from the furnaces over the fire bridges and along the internal flues to the back of the boiler where they dip into the bottom flue F which they traverse to near the front of the boiler where they divide and pass up into the side flues H. Passing along the side flues, the hot gases enter the main flue K which leads them to the chimney. The main flue shown serves a number of boilers placed side by side. By

operating dampers L, placed at the back ends of the side flues, the draught may be regulated. These dampers are operated by chains J

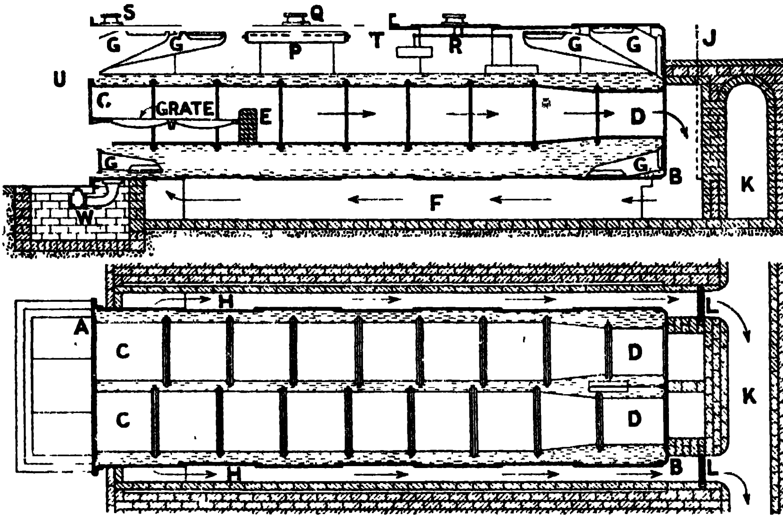


FIG. 89.--Lancashire boiler.

which pass over pulleys to the front of the boiler. In Fig. 90 the right hand half is a section through the passage between the bottom flue and the right hand side flue.

The various external flues are lined with fire-bricks and the boiler rests on fire-clay seating blocks. The bearing surface of the seating blocks on the boiler shell should not be more than 4 inches wide for the largest size of boiler.

The various details of construction are fully dealt with in the next chapter while the mountings and accessories are described in Chapters VIII and IX, but the positions of various details in Figs. 89 and 90 and not fully shown there may be indicated here. The flat ends of the shell are supported by the gusset stays G. At P is the steam collecting pipe which communicates with the stop valve which is mounted on the stand pipe Q. The low-water alarm apparatus is at R and over it is mounted a combined high-steam and low-water safety valve. A safety valve is also mounted on the stand pipe S. The manhole is at T. The feed valve U is placed at the end of an internal pipe not shown. W is the blow-off cock and elbow pipe.

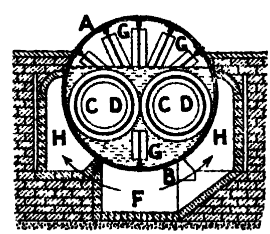


FIG. 90.—Lancashire boiler.

Air spaces are shown at the back of the fire-brick lining in the longitudinal walls in Figs. 89 and 90. These air spaces tend to prevent the cracking of the brickwork due to the expansion of the walls next to the flues. They also reduce the loss of heat conducted

through the walls. Investigations by the U.S. Geological Survey have however shown that the air spaces are much more effective in this respect if they are loosely packed with insulating material such as ash or crushed brick. This packing in the air spaces also reduces the leakage of air through the walls into the flues.

98. Marine Boilers—Scotch Type.—The Scotch type of marine boiler is the one nearly always used in the mercantile marine. This type of boiler has a cylindrical shell with flat ends. It is provided with furnaces, combustion chambers, and numerous tubes through which the furnace gases pass to the smoke-box. It is self contained in that it requires no brickwork setting, there being no external flues. The boiler may be single- or double-ended. The single-ended boiler has furnaces in one end only while the double-ended boiler has furnaces in both ends. Boilers of this type are made as small as 8 feet in diameter while the largest are nearly 18 feet in diameter. The number

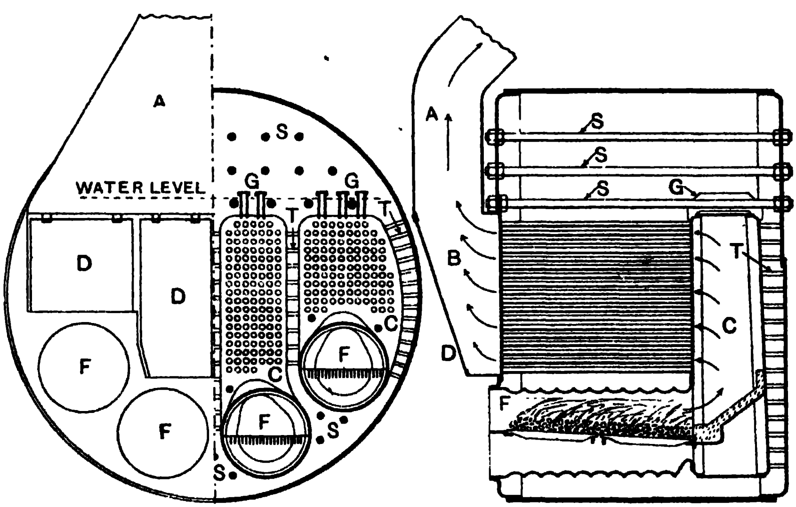


FIG. 91.—Single-ended marine boiler (Scotch type).

of furnaces in one end varies from one to four according to the diameter of the furnace and the diameter of the shell.

Fig. 91 shows a single-ended boiler having four furnaces. Each furnace F, generally corrugated, extends from the front end to a combustion chamber C from which a large number of tubes lead the furnace gases into the smoke-box B at the front end from which the gases pass through the uptake A to the funnel. The smoke-box is provided with doors D which give access to it for the purpose of cleaning the tubes and the smoke-box.

The boiler is made entirely of steel, except the tubes and the screwed stays which are generally of wrought-iron. The tubes are generally $2\frac{1}{2}$ inches in external diameter and from 7 feet 6 inches to 8 feet 3 inches long. The flat ends of the boiler above the tubes are supported by the bar stays S and the tubes support the tube plates, a number of the

tubes being specially designed to act as stays. The sides and backs of the combustion chambers are supported by the screwed stays T, while their crowns have girder stays G. The internal diameter of the furnace tube is generally from 3 feet to 3 feet 9 inches. Further particulars and illustrations of the various details are given in Chapter VII.

The two types of double-ended boiler are shown in Figs. 92 and 93. In one type (Fig. 92) each furnace has its own separate combustion chamber while in the other (Fig. 93) there is a common combustion chamber for each pair of opposite furnaces.

A single-ended boiler is shown in greater detail in Fig. 94. This boiler has a mean diameter of 16 ft. 6 in.

The following particulars of one of the twenty-one double-ended boilers of the Cunard liner *Aquitania* are of interest.[1] All of these

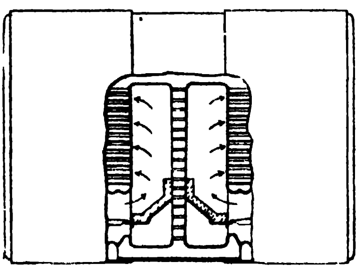


FIG. 92.

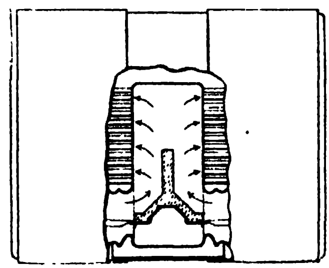


FIG. 93.

twenty-one boilers are exactly alike. Each boiler has eight furnaces each with its own separate combustion chamber. Mean diameter of shell, 17 ft. 8 in., and mean length, 22 ft. Internal diameter of furnaces (Morison type), 3 ft. 9 in. Length of fire-grate, 5 ft. 6 in. Grate area, 168·7 sq. ft. Furnace heating surface 405·6 sq. ft. Combustion chamber heating surface, 714·6 sq. ft. Tube heating surface, 5479·6 sq. ft. Total heating surface, 6599·8 sq. ft. Flue area through tubes, 25·12 sq. ft. Ratio of total heating surface to grate area, 39·1 to 1. Ratio of grate area to flue area through tubes, 6·71 to 1. Volume of steam space, 1231·7 c. ft. Working steam pressure, 195 lb. per sq. in.

99. Locomotive Boilers.—The loads which locomotives have to haul have increased greatly in recent years, so also have the speeds, and this has of course necessitated increased dimensions in the boilers and higher working pressures and it is also increasingly common to add a superheater. A good example of a recently constructed locomotive boiler for a tank locomotive, dealing with heavy traffic on the Great Central Railway, is shown in Fig. 95 and was designed by Mr. John G. Robinson, chief mechanical engineer of that railway.

The coal is introduced through the fire-hole F which is provided with a door not shown. It will be seen that the grate slopes downwards towards the front. Over the front part of the grate there is a

[1] For illustrations and further particulars of these boilers see *Engineering*, May 22, 1914.

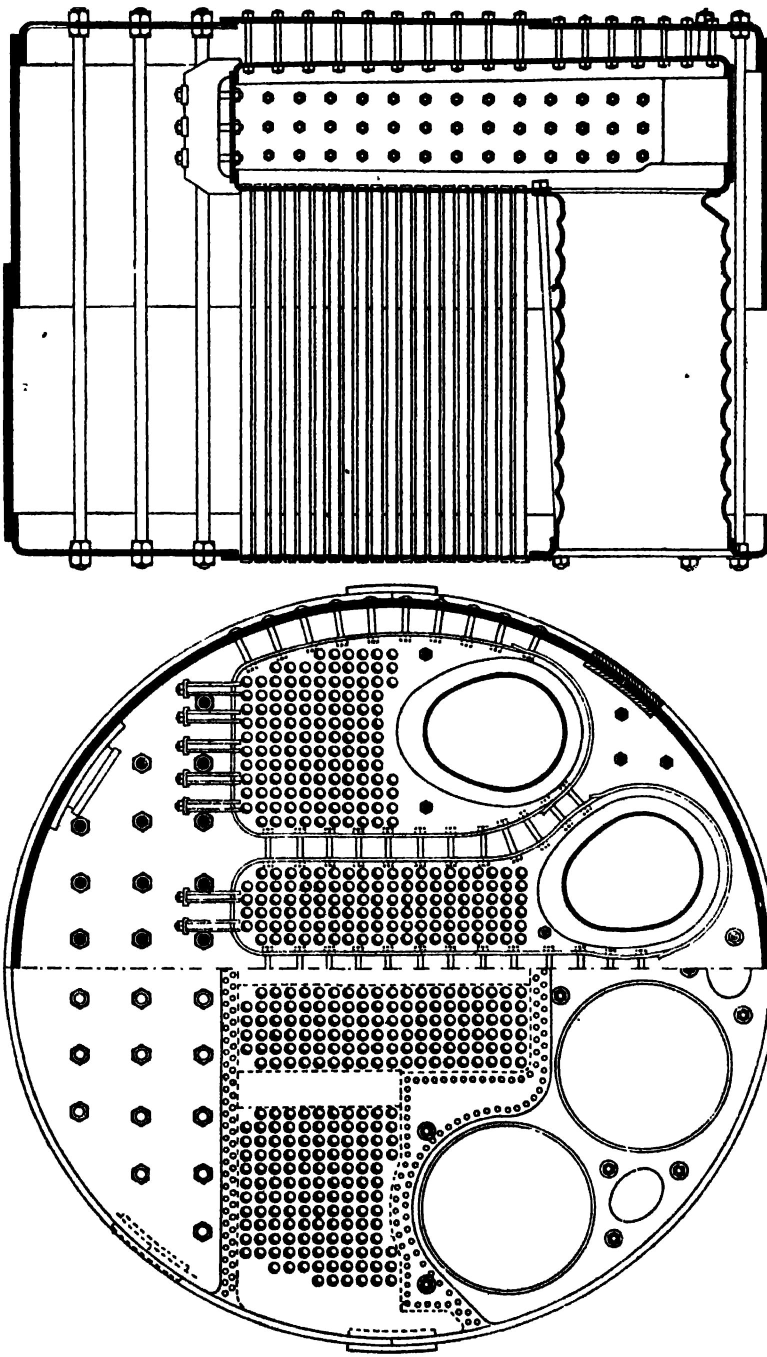


FIG 94.—Single-ended marine boiler (Scotch type).

fire-brick arch A which deflects the products of combustion, causing them to come in contact more thoroughly with the whole heating surface of the fire-box before passing through the tubes which are contained by the barrel. The tubes are expanded at their ends into the front and back tube plates. There are 157 ordinary tubes $1\frac{7}{8}$ inches in diameter and 24 tubes $5\frac{1}{4}$ inches in diameter which contain the superheater elements. After passing through the tubes the products of combustion enter the smoke-box from which they pass to the atmosphere through the chimney C.

The draught is produced by the exhaust steam from the cylinders which is discharged through the blast pipe P to the chimney. A blower is also provided for use when necessary when the steam supply to the engine is shut off. This blower, which is not shown in Fig. 95, consists of a perforated hollow ring which is placed round the mouth of the blast pipe and is supplied with steam direct from the boiler. The stop valve, called the *regulator* in locomotive practice, is placed in the dome and is operated from the cab through the rod R.

The steam pipe S leads the steam from the regulator to the saturated steam chamber of the header H of the superheater and after passing through the superheater tubes it returns to the header H, entering the superheated steam chamber from which it is lead to the cylinders through the pipes Q, one to each cylinder.

The safety valves, of which there are four, are 4 inches in diameter and they are mounted on the boiler at Y. The steam whistle is mounted at W. At the front and back of the ash-pan there are dampers D operated through rods and levers from the cab. A circular door O gives access to the smoke-box.

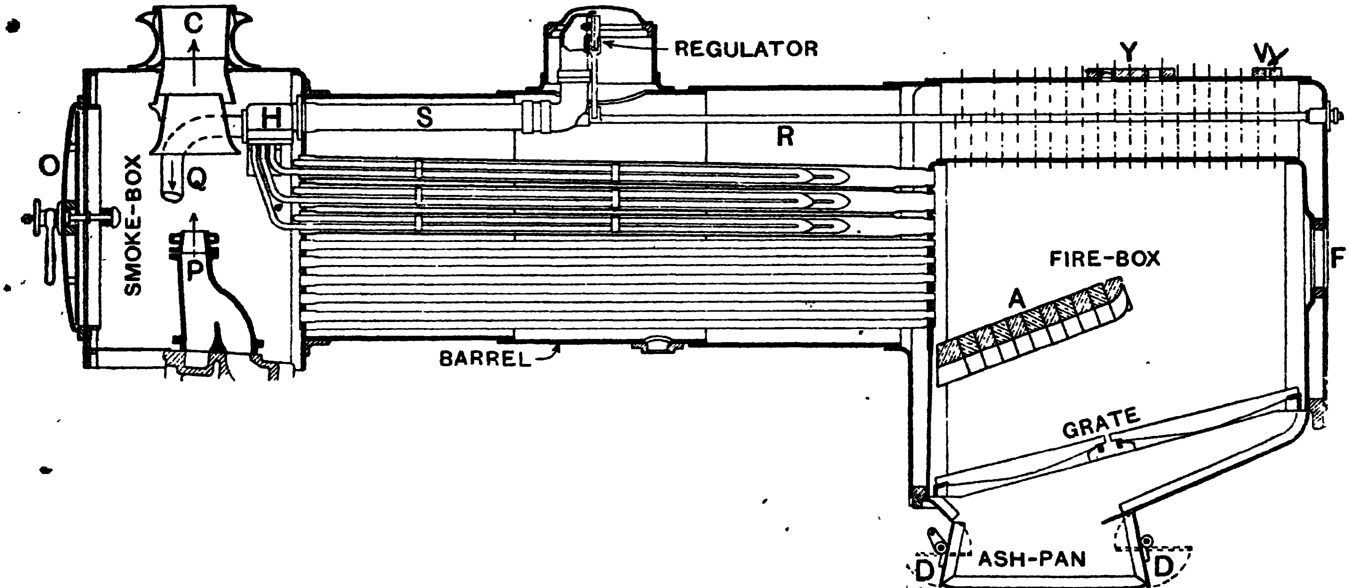
The shell surrounding the fire-box is separated from it at the sides and back, and also at the front below the barrel, by narrow water spaces. The walls of these water spaces are supported by numerous screwed stays. The tube plates are stayed within the field of the tubes by the tubes themselves. The front tube plate above the tubes is tied to the back plate of the shell by long bar stays R.

The crown of the fire-box and the roof of the shell above it, which are both flat, are supported by bar stays as in another example shown in greater detail in Fig. 96, which represents recent practice on the Midland Railway.[1]

The fire-boxes in Figs. 95 and 96 are of what is known as the *Belpaire* type now generally used in large locomotive boilers. In this type of fire-box the crown is flat and so is the roof of the shell above it. The Belpaire type of fire-box gives a greater area of water surface and a greater steam space over the fire-box crown where the heating surface is most effective and it simplifies the direct staying of the crown.

Referring to Fig. 96, this represents the fire-box of a boiler for a compound locomotive. The working pressure is 220 lb. per square inch. The mean diameter of the barrel is 4 ft. $7\frac{7}{8}$ in. and the length between the tube plates is 12 ft. $3\frac{3}{8}$ in. There are 148 ordinary tubes $1\frac{3}{4}$ inches outside diameter and 21 tubes 5 inches outside diameter to

[1] Fig. 96 has been prepared from a working drawing kindly supplied by Sir Henry Fowler, chief mechanical engineer of the Midland Railway.



Mean diameter of barrel, inside	5 ft. $0\frac{1}{2}$ in.
Length between tube plates	12 ,, $7\frac{3}{8}$,,
Fire-box, length, inside	7 ,, $7\frac{7}{16}$,,
,, least width, inside	3 ,, $4\frac{1}{8}$,,
Height from rail to centre of barrel	8 ,, 11 ,,
Height from rail to top of chimney	13 ,, $2\frac{3}{16}$,,

Heating surface, tubes	1886 square feet
,, ,, fire-box	157 ,, ,,
,, ,, superheater	304 ,, ,,
Total	1847 ,, ,,
Grate area	26 ,, ,,
Working pressure	180 lb. per sq. in.

FIG. 95.—Locomotive boiler, Great Central Railway.

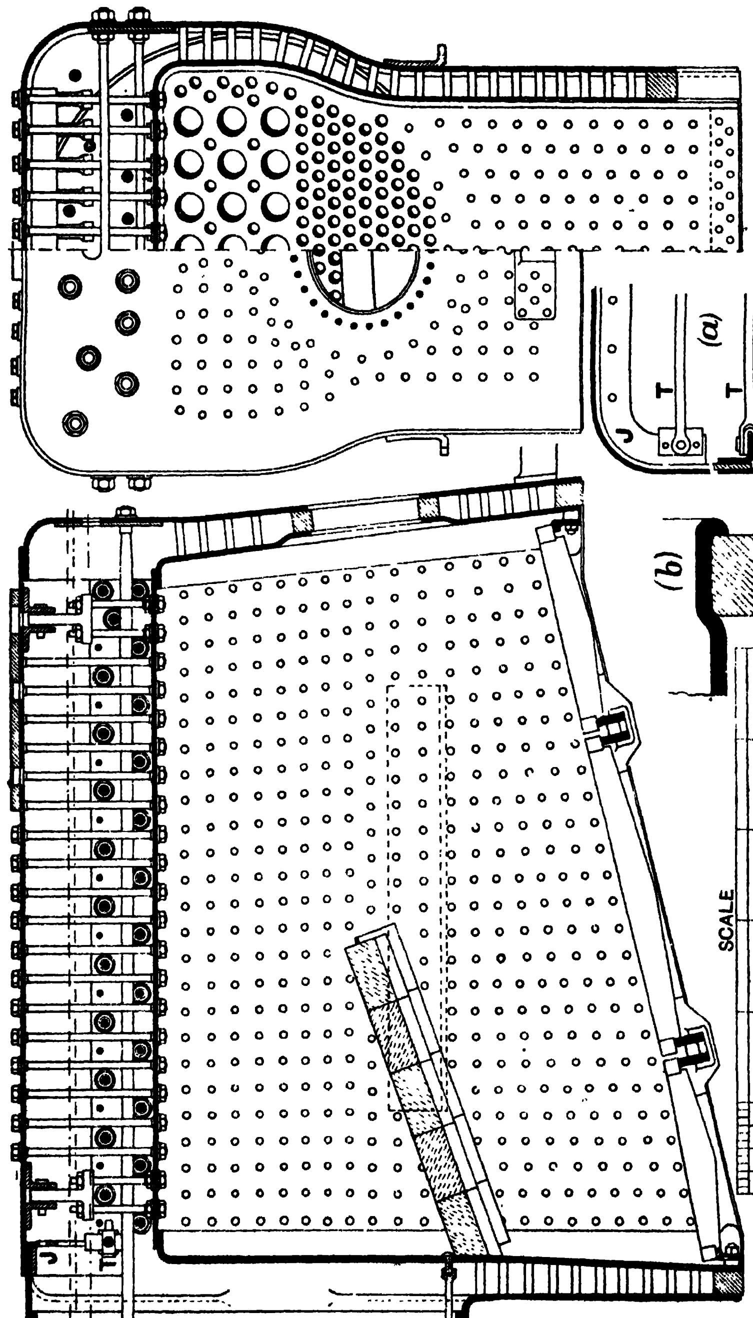


FIG. 96.—Locomotive fire-box, Midland Railway

take the superheater elements. All the tubes are of steel and are solid drawn; they are reduced in diameter at the fire-box end and enlarged at the smoke-box end.

At (*a*) is shown a separate elevation and plan of the stiffener J and stay-rod T. At (*b*) is shown, to a larger scale, part of the fire-box end of one of the larger tubes. It will be seen that this end of the tube is beaded over and that there are four shallow grooves into which the metal of the tube is forced by the tube expander.

100. Some Features of American Locomotive Boilers.—The fire-box sheets of American locomotive boilers are almost invariably made of mild steel. It is a common practice to support the fire-brick arch on tubes as shown in Fig. 97, where A is the brickwork and W the

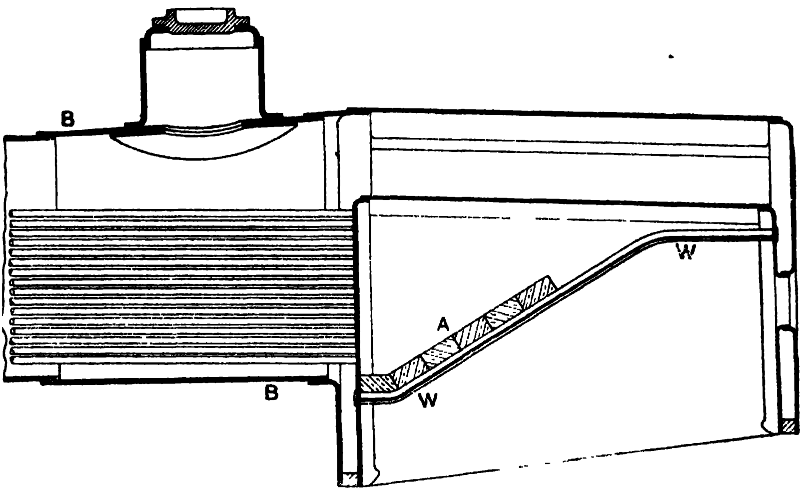


FIG. 97.—Wagon-top boiler with arch tubes in fire-box.

tubes. These tubes are 3 inches in diameter and are expanded into the front and back sheets of the fire-box and beaded over. They are placed at intervals of about a foot and they not only add effective heating surface but they greatly promote the circulation of the water. There are openings in the shell opposite to the ends of the tubes W, closed by suitable covers, so that they may be cleaned internally when necessary. In Fig. 97 the portion B of the barrel next to the fire-box is conical. From this feature the boiler is said to have a *wagon top*.

The average American locomotive boiler is larger than the average of British practice and there are on American railroads many boilers of enormous size. The following particulars of the boiler of a compound articulated locomotive constructed by the Baldwin Locomotive Works for the Erie Railroad illustrate the great developments, as regards dimensions, which have been made in American locomotive practice.[1]

The smallest diameter of the barrel is 7 ft. 10 in. The length

[1] See *Engineering*, Nov. 13, Nov. 20, Dec. 4, and Dec. 18, 1914.

between the tube plates is 24 feet. There are 326 tubes $2\frac{1}{4}$ inches in diameter and 53 tubes $5\frac{1}{2}$ inches in diameter which contain the elements of a Schmidt superheater. The grate is 10 feet long and 9 feet wide and has therefore an area of 90 square feet. Such a large grate is only possible by using a mechanical stoker.

An extension of the fire-box 4 ft. 6 in. long enters the barrel and forms an additional combustion chamber. The front of this extension forms the back tube-plate. There are six tubes extending from the bottom of the combustion chamber just mentioned to near the top of the back plate of the fire-box. These tubes carry the fire-brick arch. At the front end of the grate there is a vertical fire-brick wall which joins the brick arch at its front end.

The heating surface is made up as follows: Tubes in barrel, 6418 sq. ft. Arch tubes in fire-box, 88 sq. ft. Fire-box and combustion chamber, 380 sq. ft. Total, 6886 sq. ft. In addition the superheater presents a heating surface of 1584 sq. ft.

The working pressure is 210 lb. per square inch. There is an exhaust steam feedwater heater having 436 sq. ft. of heating surface. The boiler can be fed either by injectors or by feed pumps or by both. There are two injectors and two pumps.

101. Water-Tube Boilers.—The feature which distinguishes water-tube boilers from other multitubular boilers is that in the former the water circulates through the tubes, the hot gases being outside, while in the latter the hot gases flow through the tubes, the water being outside and contained by the boiler shell.

The water-tube boiler holds a comparatively small quantity of water, and steam may therefore be raised more rapidly than is possible with a boiler of large water capacity. Since the heat energy stored in a given volume of hot water is many times that stored in the same volume of steam at the steam temperature the energy stored in a boiler practically depends on the amount of water in it. Hence a boiler of small water capacity has a small reserve of energy. Also a temporary failure in the feed arrangements is more serious in a boiler of small water capacity than in one holding a large quantity of water.

The largest part of a water-tube boiler is the steam drum and this is much smaller than the shell of a tank boiler of the same power. The water-tube boiler is therefore simpler to transport when it is not convenient to transport it already built up.

For a given power the water-tube boiler occupies a smaller floor area than a tank boiler.

Owing to the absence of a large shell under pressure the water-tube boiler can be constructed to work at very high pressure without excessive thickness of metal.

Some interesting figures relating to the weights of cylindrical and water-tube boilers for marine purposes were given by Mr. E. M. Speakman in a paper on "The wider adoption and standardization of water-tube boilers," read before the Institution of Engineers and Shipbuilders in Scotland in Feb., 1912. The weight of steam produced, in lb. from and at 100° C. (212° F.), per ton of boiler room installation, with piping, pumps, and all accessories, might be taken as follows:—

For cylindrical boilers	150 to 200
For Babcock and Wilcox boilers (water-tube) . .	450
For Yarrow boilers (water-tube)	490

102. Babcock and Wilcox Water-Tube Boiler.—The land type of

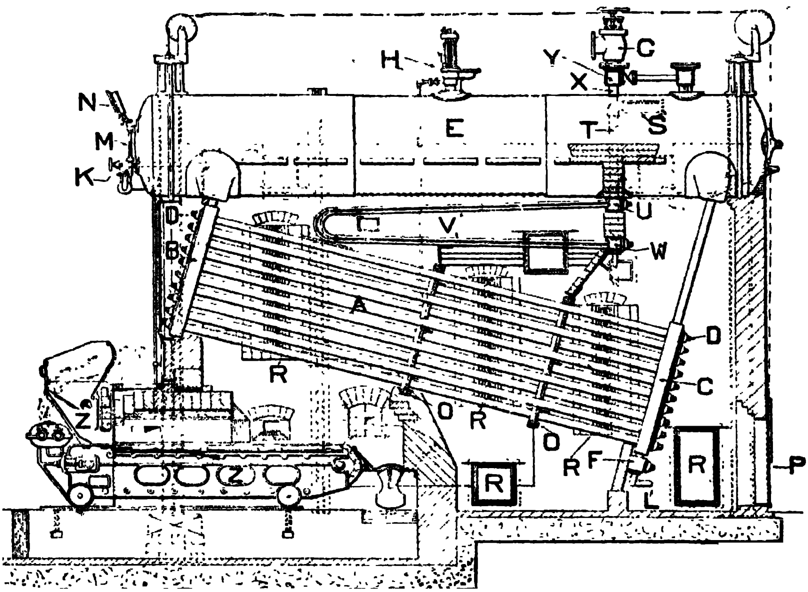


FIG. 98.—Babcock and Wilcox boiler (land type).

the Babcock and Wilcox boiler is illustrated by Fig. 98. The straight, solid drawn steel tubes A, 4 inches in diameter, which are inclined to the horizontal, connect the uptake headers B with the downtake headers C. The steam and water drum E is connected at the front end to the uptake headers by short tubes and at the back end to the downtake headers by longer tubes as shown.

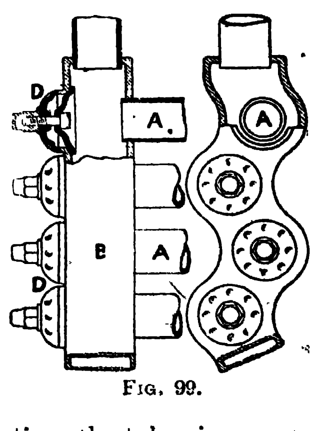


FIG. 99.

The tubular part of the boiler is made up of vertical sections, each consisting of a number of tubes and two headers, one at the front and the other at the back end. One of these headers is shown in detail in Fig. 99 from which it will be seen that the headers have a sinuous form when viewed in the direction of the tubes A, a form which enables the tubes of a section to be staggered so that in the assembled tubular sections the tubes in one row are over the spaces between the tubes in the row below. The headers

are provided with handholes placed opposite to the ends of the tubes A to permit of the cleaning or removal of these tubes when necessary. Each handhole is covered by a stamped steel cap D, which is secured by a stamped steel clamp and steel bolt and cap-nut as shown. The clamp is long enough to bridge over the handhole and narrow enough to pass through it. All the tubes are solid drawn steel tubes and they are expanded into bored holes in the headers.

The uptake and downtake tubes between the headers and the steam and water drums are connected to the latter through forged steel cross boxes which are riveted to the underside of the drum, one at each end. A perspective sketch of one of these cross boxes, which is pressed out of a single plate without weld or seam of any kind, is shown in Fig. 100.

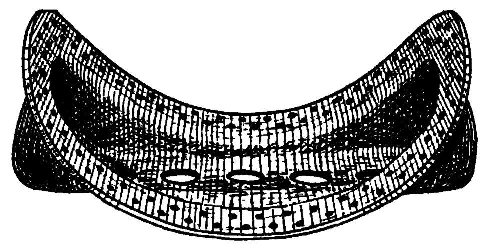


FIG. 100.

Each downtake header is joined at its bottom end to a mud box F into which any matter held in suspension in the water is, to a large extent, precipitated by reason of its greater specific gravity.

The boiler is fitted with the usual mountings, comprising, main stop valve G, safety valve H, feed valve K, blow-off valve L, water gauge M, and steam pressure gauge N.

The entire boiler, with the exception of the furnace, is suspended by wrought steel slings from wrought steel girders resting on wrought steel columns, so that the boiler can expand or contract freely without straining the brickwork which encloses the boiler and furnace. The brickwork casing is lined with firebrick.

The furnace is arranged below the tubes, and fire-brick baffles O compel the hot gases to pass upwards, then downwards, then upwards again before escaping to the chimney. The damper P, for regulating the draught, is placed in the back chamber. Doors R giving access to the interior for cleaning the tubes and removing soot are placed in the brickwork on one side.

The water circulates through the inclined tubes A into the uptake headers B then upwards into the drum E where any steam which has formed separates from the water. The water remaining moves to the back end of the drum and descends by the tubes at the back into the downtake headers C and back again into the inclined tubes A, continuing its former course until it is evaporated.

The steam superheater, which, if installed, forms an integral part of the boiler, consists of a number of solid drawn steel tubes V, bent as shown, and expanded at each end into wrought steel boxes U and W.

From the illustration (Fig. 98) it will be seen that the steam is taken from the drum E through the dry pipe S and the inlet tubes T into the superheater box U from which the steam passes through the tubes V into the lower box W. During its passage through the tubes

V the steam is superheated and is then taken from the lower box W through the outlet pipes X and the outlet cross pipe Y to the stop valve G.

When steam is being raised from a cold boiler an arrangement is provided for flooding the superheater. This consists of a connexion with the water space of the boiler and two cocks. By opening the larger of these cocks water is admitted to the superheater and fills it to the boiler water level. Any steam formed while the superheater is flooded is delivered to the drum E. When steam is raised to working pressure, and before opening the boiler on to the steam range, the large cock is closed and the small one opened, and the water flooding the superheater is drained away, a sight glass being provided to show when the draining is completed.

The boiler shown in Fig. 98 is provided with the Babcock and Wilcox mechanical chain grate stoker Z.

The marine type of the Babcock and Wilcox boiler is shown in Fig. 101. The main features of this type of the Babcock and Wilcox boiler are similar to those of the land type. The principal differences are: (1) The steam and water drum is placed transversely in the marine type instead of longitudinally. (2) The main tubes slope upwards from the front instead of downwards. (3) The circulation of the water in the inclined tubes is from the front of the boiler to the back and the uptake headers are therefore at the back and the downtake headers are at the front Also, the mud or sediment box is at the front instead of being at the back. (4) The tubes connecting the uptake headers to the drum are horizontal instead of being nearly vertical. (5) The drum in the marine boiler is fitted with baffle plates G against which the steam and water entering the drum through the tubes C from the uptake headers impinge. This causes the water to be thrown downwards while the steam separates and passes round the ends of the baffle plates into the steam space of the drum. The drum is also fitted with wash plates which prevent undue movement of the water in the drum when the ship is rolling. (6) The cap for closing the handholes opposite to the tubes in the headers is placed inside the headers instead of outside, and the handholes are therefore made of oval form instead of circular Also, the joints between the caps and the headers are not metal to metal as in the land boilers but asbestos wire woven gaskets are introduced to make the joints tight.

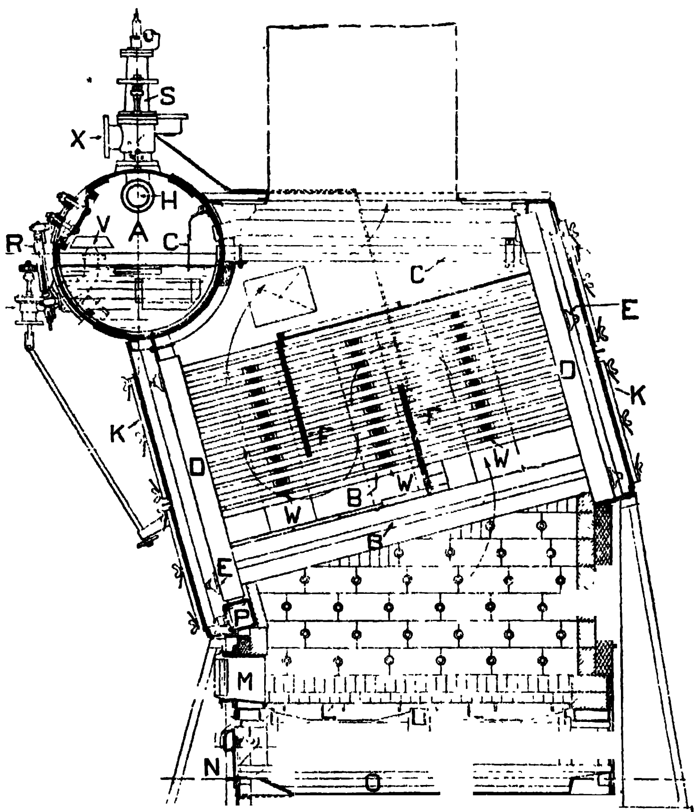
The location of the drum at the front of the boiler makes all valves and fittings accessible and tends to shorten steam pipe connections.

The furnace is either built of ordinary fire-bricks carefully fitted together, or of light fire-tiles, which are bolted to the side plates by a special arrangement. The whole boiler is encased in a special arrangement of plating fitted with fire-refractory material, which is so effective in preventing radiation of heat that the outside of the casing is kept cool.

Up to the present mechanical stokers have only been applied to a very small number of marine boilers, but the marine type of the Babcock and Wilcox boiler is often used on land in conjunction

with mechanical stokers of the Babcock and Wilcox chain grate type.

103. Stirling Water-Tube Boiler.—The following description is based on particulars kindly supplied by the Stirling Boiler Co. and



A. Steam and water drum.
B. Water tubes.
C. Return tubes.
D. Headers.
E. Handhole fittings.
F. Baffles.
G. Baffle plate.
H. Dry steam pipe.
K. Tube doors.
L. Grate
M. Fire doors.
N. Ash doors.
O. Ashpan.
P. Mud box.
R. Water gauge.
S. Safety valves.
T. Feed valve.
V. Feed nozzle.
W. Soot cleaning doors.
X. Main steam stop valve.

FIG. 101. Babcock and Wilcox boiler (marine type).

the illustration (Fig. 102) has been prepared from a working drawing also supplied by them.

There are three standard designs of Stirling boiler. In very small sizes the boilers have two steam drums and one mud drum in

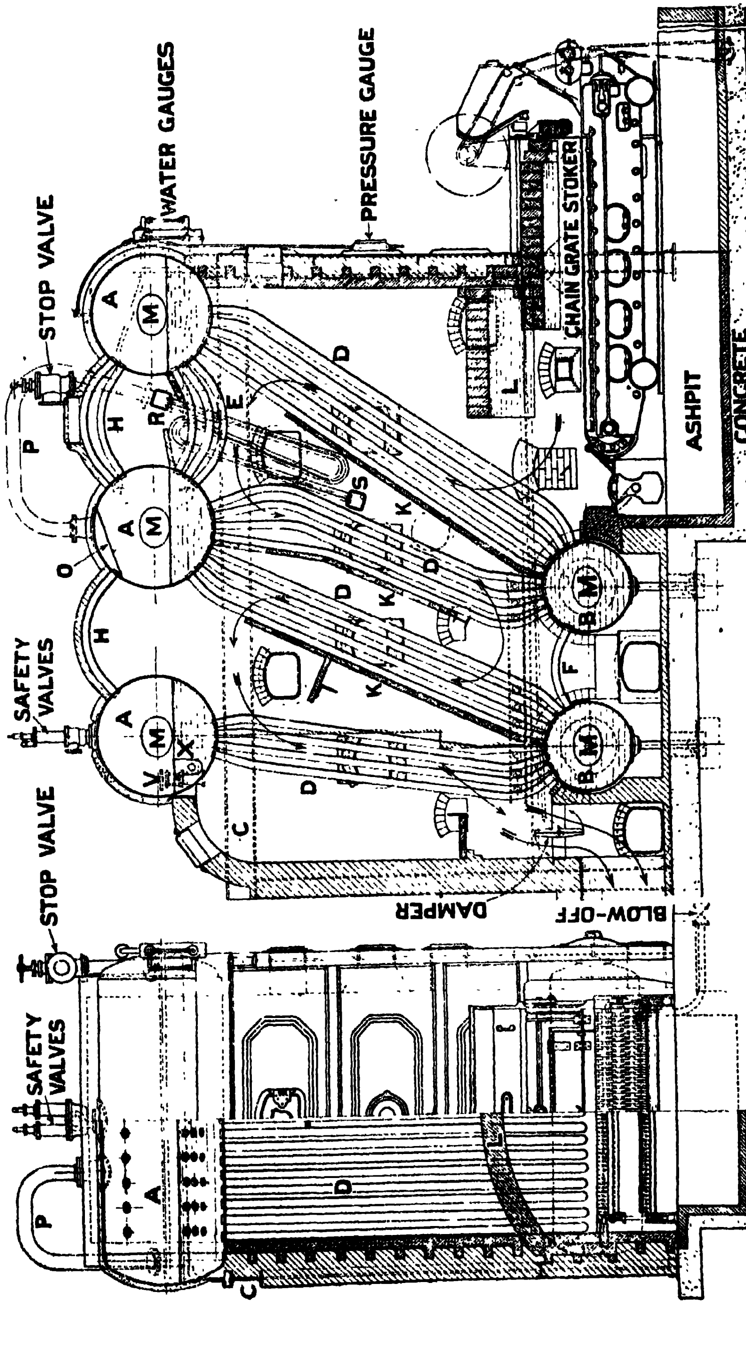


FIG. 102.—Stirling water-tube boiler.

intermediate sizes there are three steam drums and one mud drum; boilers having from 1000 to 10,000 square feet of heating surface have three steam drums and two mud drums.

The boiler illustrated by Fig. 102 has three steam drums A, and two mud drums B. The steam drums are supported by brackets carried on steel beams C which in turn rest on steel columns. These columns are built entirely independent of the brickwork, which may be removed or replaced without disturbing the boiler or its connections. The brick setting only serves the purpose of a housing to confine the heat and to provide a furnace space.

The mud drums are suspended from the steam drums by four banks of tubes D, and, not being in contact with the brick-work, are free to accommodate themselves to any movement due to expansion or contraction. Between the first and second steam drums there are water-connecting tubes E and between the mud drums there are similar tubes F. The steam spaces of the three steam drums are connected by tubes H. Fire-brick baffle tiles K arranged between the banks of tubes direct the gases into their proper course as shown by the arrows.

The steam drums in the small boilers are 3 feet in diameter and in the large boilers 4 feet in diameter. The mud drums in the smallest boilers are 2 feet 6 inches in diameter, and in all boilers having over 450 square feet of heating surface they are 3 feet in diameter. The ends of the drums are dished and flanged and each drum has a man-hole M in one end.

All the tubes are of mild steel solid drawn and they have a diameter of 3¼ inches. They are all expanded into drilled holes in the drums which they enter radially. The curvature given to the tubes allows of free expansion of the boiler when at work.

A fire-brick arch L is sprung over the grate immediately in front of the first bank of tubes, and the large triangular space above this arch between the boiler front and the first bank of tubes is available as a combustion chamber. The arch absorbs heat from the fire, becoming an incandescent radiating body which heats the air required for combustion, ignites the gases distilled from the coal, and prevents the chilling of the boiler by an inrush of cold air when the furnace door is opened.

The feed water entering the rear steam drum through the feed check valve V passes into the feed distribution box X which extends the whole length of the drum. By means of this box X the feed water is distributed over the whole width of the boiler, each tube receiving its proper proportion of feed water. The feed water passes down the rear bank of tubes, up the third bank, down the second and up the front bank into the front steam drum. Here the steam formed during the passage up the tubes disengages and passes through the upper or steam circulating tubes into the middle drum, while the unevaporated water passes through the lower or water circulating tubes into the middle drum. This water again joins the main circulation, passing down the second bank of tubes and up the front bank, continuing its former course until it is finally evaporated.

The gases flow in the opposite direction to that of the feed water.

After combustion takes place in the furnace, the gases pass up the front bank of tubes, down the second bank, up the third, and down the rear to the chimney.

In the middle of the length of the middle steam drum a compartment O is formed from which the steam is taken to the superheater. Steam formed or collected in the front steam drum has to pass to the middle steam drum and then into the rear steam drum before it reaches the compartment O. Also steam formed or collected in the middle steam drum has to pass to the rear steam drum before it reaches the

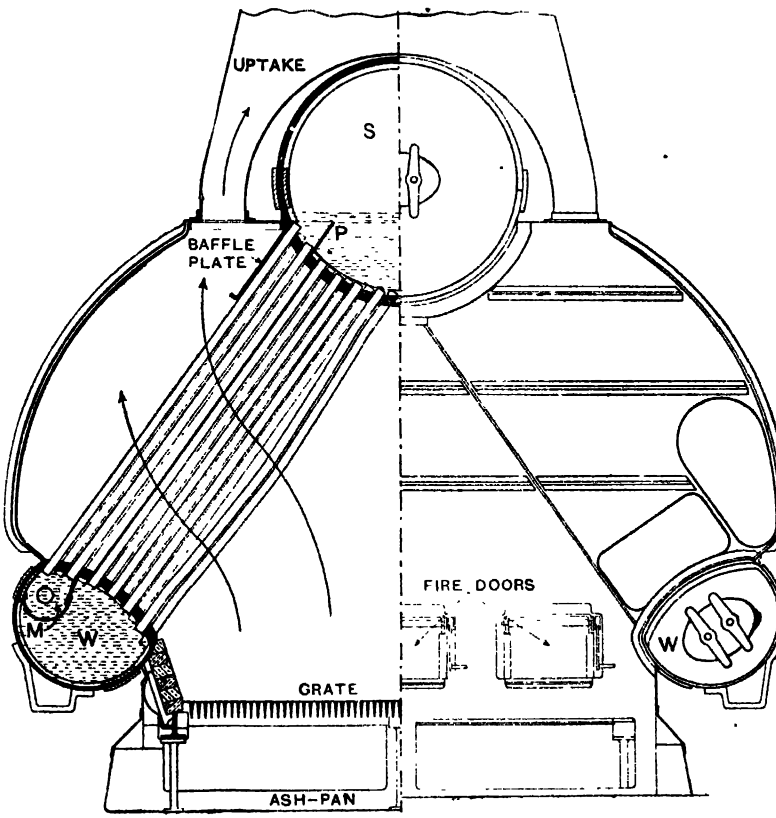


FIG. 103.—Yarrow water-tube boiler.

compartment O. The steam is led from O by the pipe P to one end of the top box R of the superheater. The top box R is connected to the bottom box S of the superheater by tubes bent zigzag as shown. There is a partition in the middle of the length of the box R and the steam entering R from O fills one-half of R and passes to S by way of one half of the superheater tubes, returning from S to the other half of R by way of the other half of the superheater tubes and then into the pipe leading to the stop-valve.

Opposite the middle of each bank of tubes there is a door in the

side wall, called a sooting door, through which a jet of steam from a steam hose can be introduced to clean the outside of the tubes. There are also other doors shown which give access for cleaning, inspection, and repairs.

104. Yarrow Water-Tube Boiler.—A half front elevation and a half transverse section of the Yarrow water-tube boiler are given in Fig. 103. This boiler is extensively used in warships of all sizes. The main parts of this boiler are: the water pockets W, the steam drum S, and the tubes connecting the former with the latter. Formerly all the tubes were quite straight but it is now common to give a slight curvature to the first two rows on the fire side as shown in Fig. 103. This curvature gives the tubes a slight flexibility and tends to maintain their tightness in the tube plates.

In the boilers for large ships the tubes are generally $1\frac{3}{4}$ inches in diameter and in the boilers for torpedo boats and other small craft they are generally 1 inch or $1\frac{1}{4}$ inches in diameter.

A compartment M is formed in each water pocket. The feed water is delivered into these compartments which compel the water first to traverse the outer two or three rows of tubes which then act as feed water heaters. Longitudinal division plates P ensure the mixing of the feed water with the bulk of the water in the steam drum after passing through the feed-heater tubes.

At a meeting of the Institution of Naval Architects Mr. Yarrow described the method of replacing tubes in his boiler as follows.

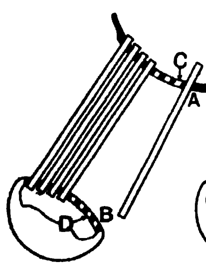


FIG. 104.

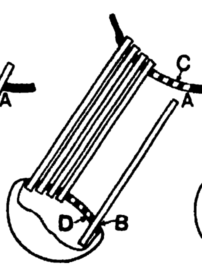


FIG. 105.

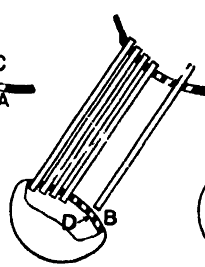


FIG. 106.

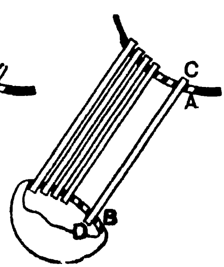
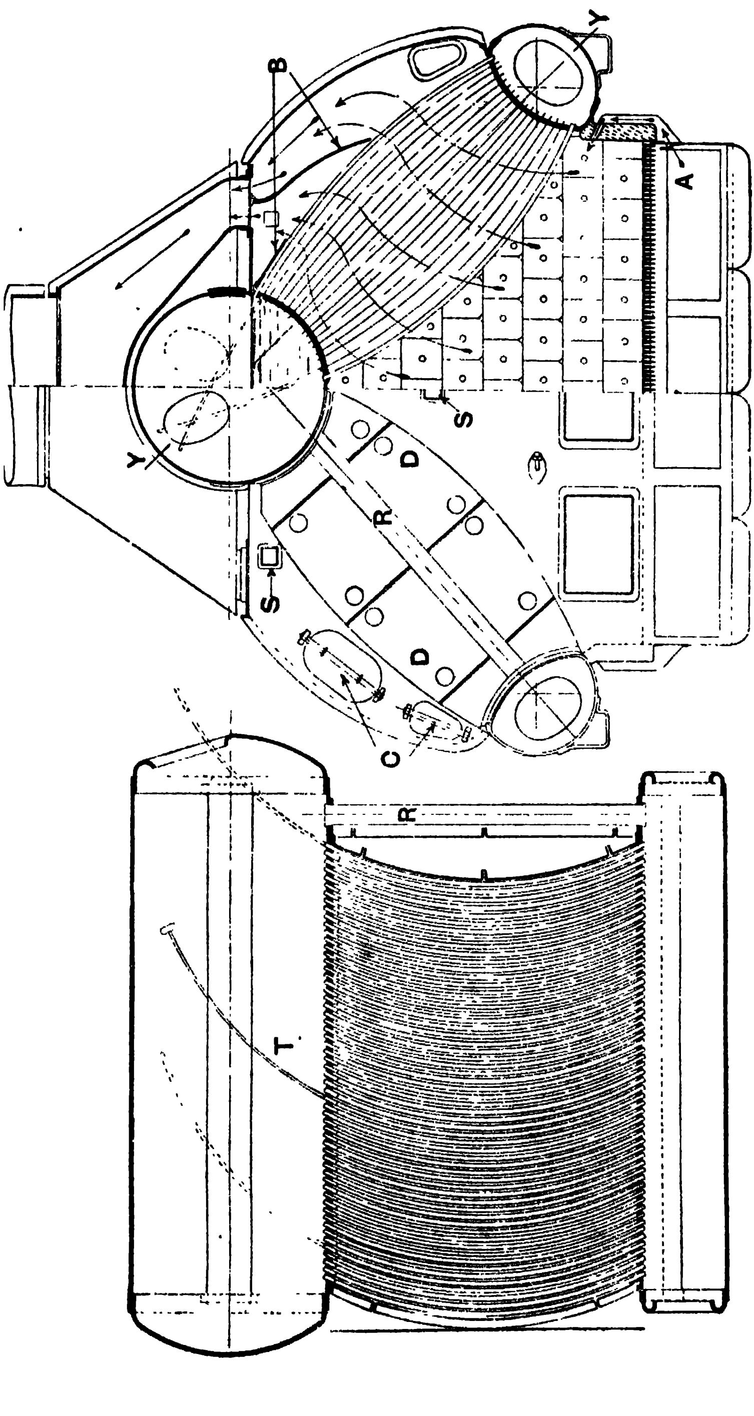


FIG. 107.

Referring to Figs. 104 to 107 it will be seen that a tube can be passed along from hole to hole very readily to any part of the tube plate. The top end of the tube is passed into the first of the holes A in the steam drum (Fig. 104), it is then passed down into and through the first hole B in the water pocket (Fig. 105), then up again into the next hole C (Fig. 106) in the steam drum, then brought down again into the next hole D in the lower tube plate (Fig. 107), and so on into further holes if necessary. The tube can be, as it were, stepped from hole to hole, without any difficulty whatever, to any required spot. By reversing the action any tube can be taken out, removing in the first instance the tubes which are in front of it.

105. White-Forster Water-Tube Boiler.—The main features of the White-Forster water-tube boiler are similar to those of the Yarrow boiler, but it will be seen by reference to Fig. 108 that the tubes are



Y-Y FRONT VIEW

FIG White-Forste ıer-tube boiler.

all curved. All the tubes have the same curvature and this curvature is sufficient to determine the direction of movement due to expansion and prevents distortion when the rate of working is high. The tubes are arranged in position in a transverse section, like the staves in a section of a barrel, and the continuation of the line of curvature of each tube, passing through the steam drum in line with the end manhole, allows each tube to be inserted or withdrawn through this manhole, so that it is possible to withdraw any tube without disturbing any of the others. As all tubes have the same curvature they can be readily cleaned internally by a tube brush having a rigid handle curved to the same radius as shown at T.

The boiler and its casing are constructed entirely of mild steel. Large downtake tubes R are fitted at one end. Baffles B are fitted to divide the uptake into two or more compartments in such a manner that the gases are directed equally over the whole tube surface and at right angles to it. Air holes A are arranged at the sides and ends for the admission of air above the grate. Doors C for soot cleaning, doors D for tube cleaning, and sight holes S are fitted.

Like the Yarrow boiler the White-Forster boiler is largely used in warships, especially in torpedo boats, torpedo boat destroyers, and vessels of light draught where lightness, combined with high power. is important.

106. The Bonecourt Surface Combustion Boiler.—If a combustible gas, mixed with sufficient air for its complete combustion, be directed against a suitable porous material, which has been heated to incandescence, it is found that the gas burns *without flame* at the surface of the porous material, and a very high temperature is produced. The combustion in this case is called *surface combustion*. The phenomenon of surface combustion was first discovered by Sir Humphrey Davy in 1817, but its successful application to industrial purposes is mainly the result of the investigations of Professor W. A. Bone which began in 1902. In the practical applications of surface combustion Professor Bone has been associated with Mr. C. D. McCourt and their steam boiler, now to be described, is known as the Bonecourt surface combustion boiler.

The plant illustrated by Fig. 109 comprises a boiler, a feed water heater, a fan to produce the necessary draught, and an electric motor to drive the fan. At (*a*) is shown, to a larger scale, the front end of one of the boiler tubes and details connected with it.

The boiler shell which is cylindrical and 3 feet to 4 feet long is traversed by a considerable number of boiler tubes which contain, at their front ends, fireclay plugs P, each of which has a central hole for the admission of a combustible mixture of gas and air. The boiler tubes are packed with granular or fragmentary refractory material such as crushed fire-brick which will pass a 1-inch mesh but be rejected by a ½-inch mesh. The refractory material is kept in the tubes at the back ends by grids.

Attached to the front of the boiler there is a gas-box to which the combustible gas is admitted under suitable pressure, which in the case of coal gas is 2 inches water gauge, the supply being regulated by the valve shown. Openings are provided in the gas-box through

which pass short mixing tubes M which are joined by reducing sockets to smaller tubes R which enter the fireclay plugs in the boiler tubes. Opposite the outer end of each of the mixing tubes is a nozzle N supplied with gas from the gas-box and controlled by a valve as shown more clearly at (*a*). In the general view in Fig. 109 only two of these nozzles with their valves are shown. Any number of the boiler tubes may be put out of action by closing the valves of the nozzles which feed them with gas, and by screwing iron caps over the front ends of their mixing tubes to prevent the cold air being drawn through them.

The smoke-box at the back end of the boiler is provided with doors, not shown in Fig. 109, to give access to the boiler tubes for the purpose of charging them with refractory material.

The products of combustion pass from the smoke-box to the feed-

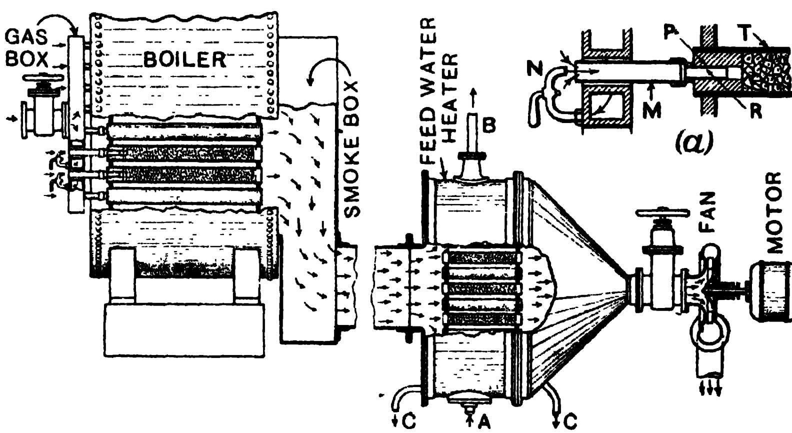


Fig. 109.—Bonecourt surface combustion boiler.

water heater, which is of cylindrical form and contains tubes charged with refractory material similarly to the boiler tubes. The products of combustion pass through the tubes of the feed-water heater into a conical duct which leads them to the fan which delivers them into the chimney. The conical duct between the heater and the fan may be removed to give access to the tubes of the heater for the purpose of charging them with refractory material.

Owing to the low temperature to which the products of combustion are cooled in passing through the tubes of the feed-water heater, condensation of water vapour may take place. The resulting water is drained off by the pipes CC which dip into water seals and which are of such length that air will not be drawn up through them by the draught produced by the fan. The cold feed-water enters the heater at A and leaves it at B.

The mixing tubes M are of such length and cross section as to cause the combustible gas and the air drawn into them under the action of the fan to mix completely before entering the boiler tubes. The smaller tubes R connected to the mixing tubes M are of such a cross section as to cause the homogeneous and explosive mixture of gas

and air to be drawn through them at a speed considerably in excess of the speed of back firing of the mixture, so that no combustion of the explosive mixture can take place before it reaches the granular bed of material in the boiler tubes.

With the arrangements described, and when employing coal gas, the inventors have found that the tubes R may conveniently be of $\frac{5}{8}$-inch bore and about $3\frac{1}{2}$ inches long, while the tubes M may be of 1-inch bore and about 8 inches long. With these dimensions a suction at the fan amounting to 15 inches of water is suitable. An important function of the fireclay plugs P is to protect the joints of the boiler tubes and front tube plate from overheating.

In starting up the boiler the fan is first set going and the gas turned on and ignited at each nozzle N so that the resulting flames extend through the mixing tubes M on to the granular material at the front ends of the boiler tubes. As soon as the granular material near the fireclay plugs P becomes sufficiently incandescent each gas jet N is turned off and then immediately turned on again with the object of extinguishing the flames and of causing a mixture of gas with air sufficient for its complete combustion to be drawn on to the incandescent granular material in the boiler tubes where the combustion now takes place.

Tests of the Bonecourt boiler and feed-water heater of the type described have shown an efficiency of over 90 per cent. after deducting the power required by the fan. This high efficiency is mainly due to the complete combustion of the fuel with a minimum of excess of air and to a low temperature of the chimney gases, about 95° C. (203° F.).

An equivalent evaporation as high as 35 lb. of water from and at 100° C. per square foot of boiler tube heating surface has been obtained in this boiler.

CHAPTER VII

STEAM BOILER DETAILS

107. Riveted Joints in Boiler Shells.—In good boiler work the edges of the plates are planed, generally to a slight bevel as shown in Fig. 110, which also shows the difference between caulking A and fullering B, operations on riveted joints to ensure steam tightness. Of these two operations fullering is preferable to caulking.

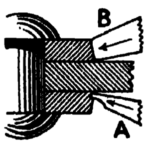


FIG. 110.

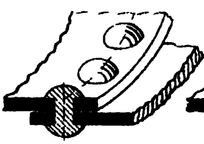


FIG. 111.

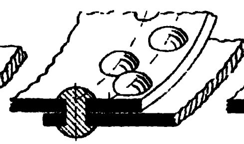


FIG. 112.

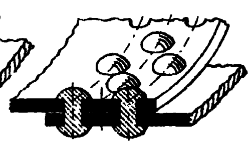


FIG. 113.

The joints for the circumferential seams are nearly always lap joints, single, double, or treble riveted, as shown in Figs. 111, 112, and 113.

All holes for rivets are drilled after the plates are bent and put together.

For the longitudinal seams butt joints with cover or butt straps outside and inside are used. Fig. 114 shows a double riveted butt

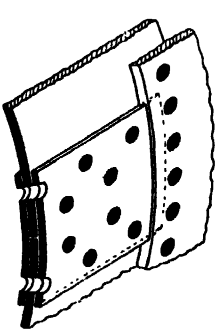


FIG. 114.

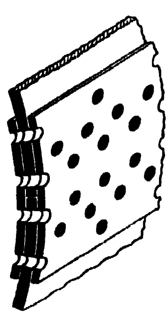


FIG. 115.

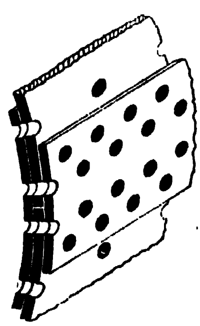


FIG. 116.

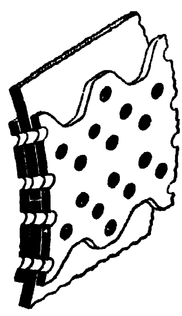


FIG. 117.

joint with two cover straps for a longitudinal seam. This illustration also shows the intersection of a longitudinal joint with a circumferential joint. It will be seen that the outer butt strap is thinned down at the end where it is tucked under the next ring plate which has a

corresponding recess planed in it, an arrangement adopted to secure steam tightness at the intersection of the two joints.

The treble riveted butt joint shown in Fig. 115 is a very common one for the longitudinal seams of large boiler shells with high steam pressures. In this joint the rivets in the outer rows have twice the pitch of those in the other rows. This wide pitch of the rivets adjacent to the edges of the butt straps tends to prevent close contact between the butt straps and the shell at the middle of the spaces between these rivets, and caulking or fullering may not be effective in making a steam-tight joint. By making the outer butt strap narrower, as shown in Fig. 116, the joint may be efficiently caulked or fullered on the outside. Scolloping the edges of the butt straps as shown in Fig. 117 is another way of getting over the difficulty connected with a wide rivet pitch. This scolloping brings the caulking edges of the butt straps nearer to the rivets.

The principal ways of connecting the end plates to the cylindrical shell are shown in Figs. 118 to 121, where E is the end plate and S

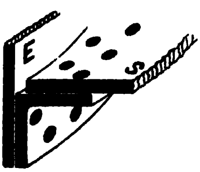


FIG. 118.

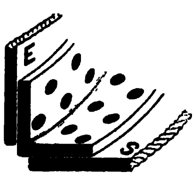


FIG. 119.

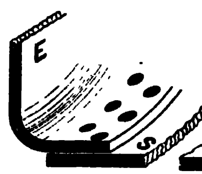


FIG. 120.

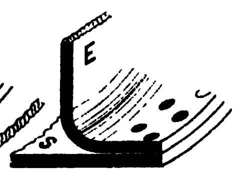


FIG. 121.

the shell plate. The arrangement shown in Fig. 118 is largely used for the front ends of Cornish and Lancashire boilers. Here a welded ring of angle section is riveted to the shell and to the end plate, the angle ring being on the outside of the shell. This arrangement is quite unsuitable for the back end of a Cornish or Lancashire boiler since the projecting angle ring and end plate would rapidly waste away by furnace gases passing over them. This objection to the angle ring connection does not apply if the angle ring is placed on the inside of the shell as shown in Fig. 119, but most commonly the back end plate of a Cornish or Lancashire boiler is flanged as shown in Fig. 120, and the shell is riveted to the flange which is inside the boiler. Except for the front ends of Cornish and Lancashire boilers the flanging of the plates has superseded the use of angle rings for the connection of end plates to the shell in all classes of boilers. With a flanged plate there is only one seam to rivet while with an angle ring there are two.

In the arrangement shown in Fig. 121 the end plate is flanged outwards. The only advantage which this design has is that it may always be machine riveted. If both ends of a boiler are flanged inwards as in Fig. 120 there is one circumferential seam which must be riveted by hand. One objection to flanging the end plate outwards is that the boiler shell is heavier for the same internal volume. Another objection is that the joint can only be caulked on the outside. Like the external angle ring connection in Fig. 118 the outward flanged

end plate in Fig. 121 is quite unsuitable for the back end of a Cornish or Lancashire boiler and for the same reason.

108. Furnace Tubes.—Boilers of the Cornish, Lancashire, Scotch marine, and some other types contain one or more *furnace tubes*, each grate being in the front portion of one of these tubes. These furnace tubes have an internal diameter varying from about 2 feet in the smallest size to about 4 feet in the largest. The furnace tubes are either plain cylinders flanged outwards at the ends or they are corrugated.

Plain furnace tubes are made up of short flanged units not exceeding 4 feet in length which are joined together as shown in Fig. 122, which represents the *Adamson flanged seam.* Here two flanged ends are riveted together with a *caulking ring* R between them. The caulking ring enables the joint to be thoroughly caulked inside as well as outside and it also adds to the strength of the tube to resist collapse. It will be observed that all the rivet heads are in the water space. The flanged seams give a certain amount of longitudinal flexibility to the furnace tubes, which reduces the straining actions due to unequal expansion of the tubes and boiler shell.

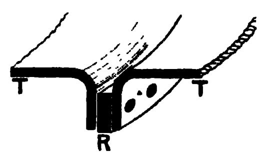


Fig. 122.

Four different forms of corrugations for corrugated furnace tubes are shown in Figs. 123 to 126. *Fox's corrugated furnace tube*

Fig. 123.—Fox's corrugations.

Fig. 124.—Morison's corrugations.


Fig. 125.—Deighton's corrugations.


Fig. 126.—Suspension bulb corrugations.

(Fig. 123) was the first of the corrugated furnaces. This was followed by *Morison's* or the *suspension furnace tube* (Fig. 124). Then came *Deighton's* (Fig. 125). The most recent successful design is the *suspension bulb furnace tube* made by the Leeds Forge Co. (Fig. 126).

Corrugated furnace tubes have the advantage of longitudinal flexibility, and greater resistance to collapse than plain tubes *especially when they become overheated* due to scale or a film of oil deposited on them from impure feed water.

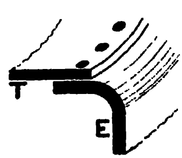


Fig. 127.

Fig. 128.

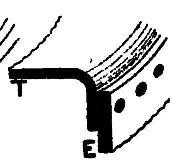


Fig. 129.

Figs. 127, 128, and 129 show flanged joints between furnace tubes and the end plates, T being a furnace tube and E an end plate. In Figs. 127 and 128 the end plate is flanged while in Fig. 129 the furnace

tube is flanged. In the joint shown in Fig. 128 the end plate is flanged outwards, a design which may be used on the front end but not on the back end. The designs shown in Figs. 127 and 129 may be used at either end of the furnace tube. Angle rings are now seldom used for connecting furnace tubes to the plates at their ends.

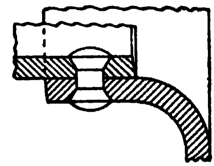


FIG. 130.

Fig. 130 shows a good form of rivet for the joints at the ends of furnace tubes, and for the seams in combustion chambers and parts exposed to flame.

Two methods of connecting the back end of the furnace tube to the front of the combustion chamber, in boilers of the Scotch marine type, are illustrated pictorially by Figs. 131 and 132. The joints are shown as viewed from the combustion chamber in each case. In Fig. 131 the furnace tube is flanged at the top only, while in Fig. 132 it is flanged all the way round. In Fig. 132 a neck is formed on the tube while it is being flanged, a corrugation of varying depth being formed adjacent to the

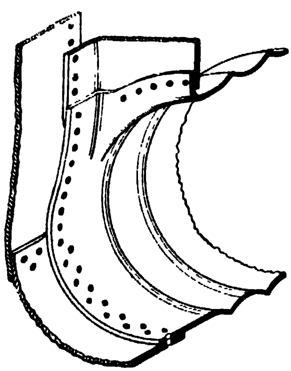


FIG. 131.

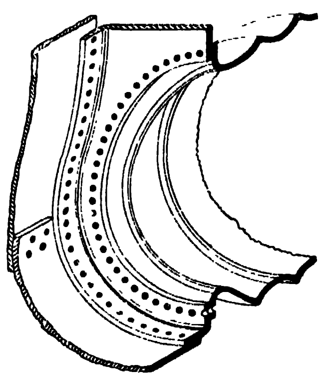


FIG. 132.

flange, the depth being greatest at the bottom. This makes the outline of the flange of the tube oval-shaped but narrowing more rapidly towards the top than the bottom, and it also makes the whole flange small enough to pass through the hole for the furnace tube in the front end plate of the boiler.

109. Galloway Tubes.—At one time it was the general practice to fit into the furnace tubes of Cornish and Lancashire boilers a number of cross tubes of conical form called *Galloway tubes* from the name of their inventor. These tubes were generally flanged and riveted into the furnace tube as shown in Fig. 133, but occasionally they were welded in.

It was claimed for these tubes that they added greatly to the strength of the furnace tube, that they provided very effective additional heating surface, and that they promoted a better circulation of the water. The first claim is now considered doubtful and the few experiments which have been made to test the last have not been very

conclusive, although one would expect that the cross tube would improve the circulation. In any case these tubes add considerably to the cost of the boiler and if scale should form in them, which it is liable to do, they are very likely to become overheated and damaged.

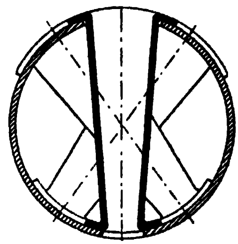


FIG. 133.

Modern boilers of the Cornish and Lancashire types are not fitted with these tubes. Additional heating surface is better obtained by means of a superheater placed in the downtake at the back of the boiler together with an economizer placed between the boiler and the chimney.

110. Boiler Stays.—In steam boilers plates which are flat or only slightly curved generally require to be supported by *stays*. The principal kinds of stays in use are: (1) Direct stays which are usually round bars placed at right angles to the plates supported by them; (2) diagonal and gusset stays used for supporting one plate by tying it to another at right angles to it; (3) girder stays which are placed edgewise on the plate to be supported and bolted to it at intervals.

Direct stays.—Bar stays for supporting one end plate of a boiler shell from the other end plate are shown in Fig. 134. The ends of the bar are screwed to receive two nuts between which the end

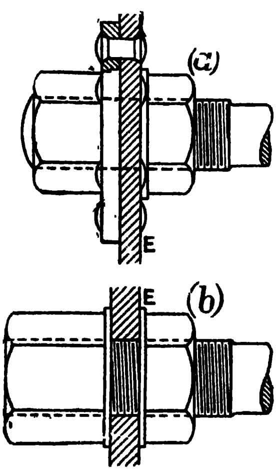


FIG. 134.

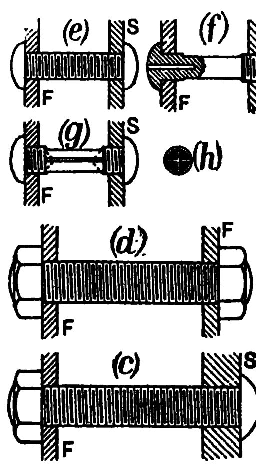


FIG 135.

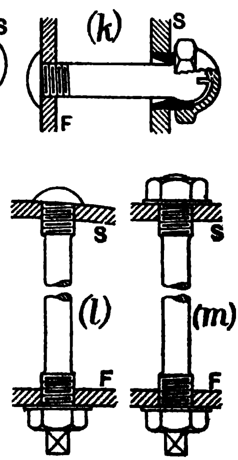


FIG. 136.

plate E is locked. At (*a*) the end plate is stiffened by a large thick washer riveted to it. These bar stays are now generally made of steel and preferably the screwed part is enlarged by staving up the bar in a hydraulic press, the stay being afterwards annealed. To reduce the amount of enlargement necessary finer screw threads are now generally used with a nut of extra depth on the outside as shown at (*b*), which is the design of bar stay used in the boilers

of the Cunard liner *Aquitania*. These stays are $2\frac{7}{8}$ inches in diameter in the body enlarged to $3\frac{1}{8}$ inches at the screw which has eight threads per inch. Bar stays such as are shown in Fig. 134 are not screwed into the plates.

The fire-boxes or combustion chambers of locomotive and marine boilers are supported by screwed stays of which examples are given in Figs. 135 and 136, where F is a fire-box or combustion chamber sheet and S the outer shell. These stays are called screwed stays because they are screwed into the plates which they support.

Marine boiler screwed stays are made of wrought iron or steel and vary in diameter from $1\frac{1}{4}$ inches to $1\frac{5}{8}$ inches and are placed at intervals of 6 inches to 9 inches. They are provided with nuts inside the combustion chambers as at F in (*c*) and (*d*), but in the wing boxes where the outer ends of the stays pass through the shell of the boiler the outer ends are riveted over as at S in (*c*).

In British locomotive practice the short screwed stays are nearly always made of copper and after being screwed into the plates they are riveted over as shown at (*e*), (*f*), and (*g*). These stays are generally about 1 inch in diameter with 11 or 12 threads per inch and are placed at intervals of about 4 inches. It is a common practice to have these stays screwed at the ends only, the middle part being turned down as shown at (*f*) and (*g*). This makes the stay slightly more flexible. Greater transverse flexibility is obtained without much reduction of cross sectional area by making thin saw cuts in the stays as shown at (*g*) and (*h*). Tate's flexible stay is shown at (*k*); this has been successfully used in American locomotive practice where the screwed stays are made of steel. Transverse flexibility in a screwed stay prevents its fracture due to unequal expansion of the fire-box and shell plates. The fracture of a screwed stay may be detected by the escape of steam through the small axial hole which is sometimes drilled in the stay as shown at (*f*).

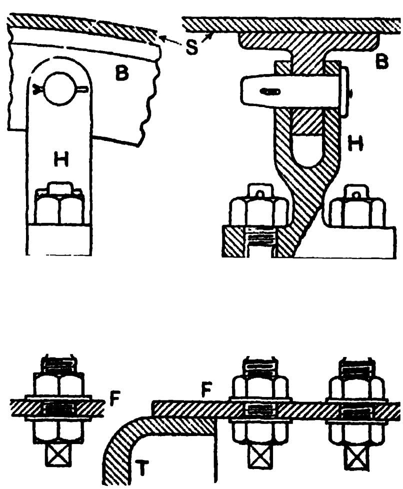


Fig. 137.

Screwed roof stays for locomotive fire boxes are shown at (*l*) and (*m*), Fig. 136. These direct stays now largely supersede the girder stays formerly used for supporting the crowns of locomotive fire-boxes. With these stays it is however necessary to allow for the upward expansion of the tube plate when the fire is first lighted. Fig. 137 shows a common form of expansion roof stay used near the tube plate. S is the outer shell, F the fire-box crown plate, and T the tube plate. Bars B of T-section are riveted to the inside of the shell and hangers H are suspended from these by pins as shown. C are stays screwed into

the fire-box crown and provided with nuts and copper washers, but they are free to move upwards in the hanger. There are generally two transverse rows of these double stays at the front or tube plate end of the fire-box crown.

Diagonal and gusset stays.—The flat end of a boiler shell may be supported from the cylindrical part of the shell by diagonal bar stays of which an example is shown at (*a*) in Fig. 138. Instead of having pin joints at both ends of the bar the end connected to the cylindrical part of the shell S is frequently riveted to it as shown at (*b*).

A gusset stay is a diagonal stay in which a plate is used instead of a bar. An example of a gusset stay is shown in Fig. 139. The

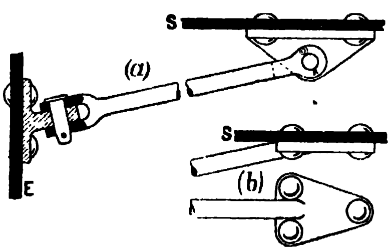


FIG. 138.

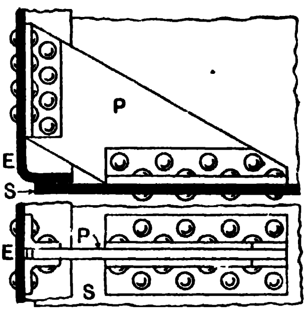


FIG. 139.

gusset plate P is connected to the cylindrical shell S and to the end plate E by angle bars and rivets as shown.

Girder Stays.—The crowns of the combustion chambers of marine boilers of the Scotch type are usually supported by girder stays of the form shown in Fig. 140. These girders, which are made of wrought iron or mild steel, are supported at their ends by the vertical front and

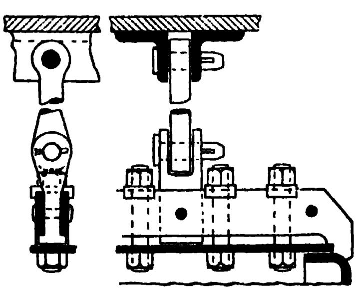


FIG. 140.

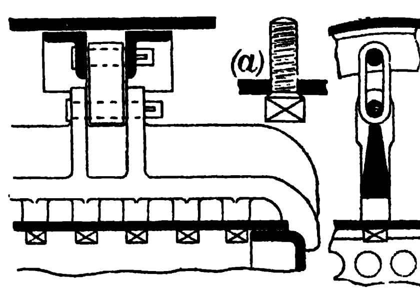


FIG. 141.

back plates of the combustion chamber, and the crown plate is supported from the girders by bolts which are screwed into the crown plate and provided with nuts as shown. In large combustion chambers, which occur in double-ended boilers in which furnaces in opposite ends lead into the same chamber, it is usual to relieve the load on the plates

which support the girders by means of sling stays hanging from the shell of the boiler as shown in Fig. 140. Combustion chambers which serve one furnace only, being much narrower, their girder stays do not require the addition of sling stays.

Fig. 141 shows a girder stay for a locomotive fire-box. The girder in this example is a steel casting and is provided with two sling stays. The method of attaching the fire-box crown to the girders is shown in detail at (*a*).

When sling stays are provided provision is made for the upward expansion of the combustion chamber or fire-box when the fire is started. The hanging stay in Fig. 140 has the holes for the pins slightly elongated so that there is a clearance of about $\frac{1}{16}$ inch below the top pin and above the bottom pin when the boiler is entirely cold or entirely hot. In Fig. 141 the link form of the sling gives ample provision for expansion.

In large modern locomotives, as has already been mentioned, direct stays have practically superseded girder stays on account of the latter requiring to be large and clumsy and in consequence obstructing the free circulation of the water over the fire-box crown.

111. Manholes and Handholes in Boiler Shells.—To give access for cleaning and inspection, boiler shells must be provided with openings having covers which may be readily removed or replaced. In all except very small shells there should be at least one opening large enough to allow a man to pass through it; such an opening is called a

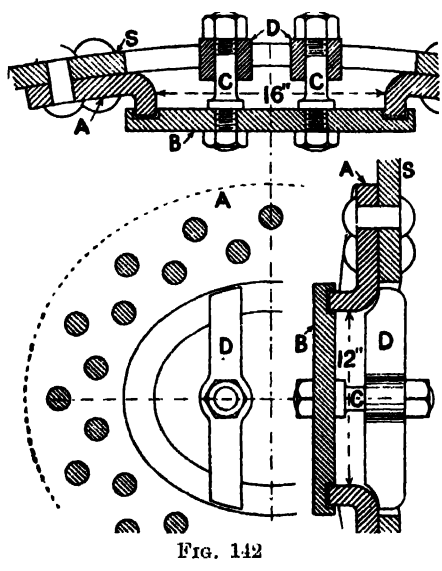


FIG. 142

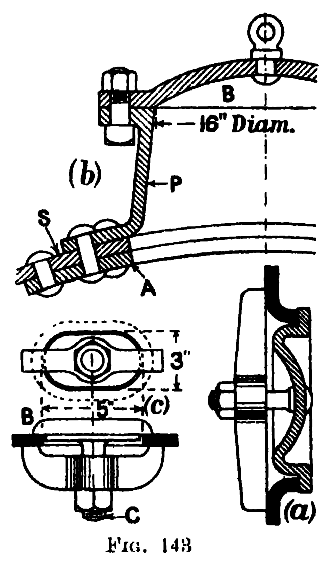


FIG. 143

manhole. Oval-shaped manholes are generally 16 inches by 12 inches and when these are in the cylindrical part of the shell the shorter axis of the hole should be placed parallel to the axis of the shell. The reason for this disposition of the shorter axis of the hole is that a

longitudinal section of the shell being weaker than a circumferential section the latter can better afford to be weakened.

Fig. 142 shows an oval-shaped manhole in the shell of a large marine boiler. S is the boiler shell which is reinforced round the opening by the plate A which is flanged inwards. The edge of this flange is faced to form a seat for the cover B which is secured by the bolts C and bridge bars D as shown.

If the manhole is made in the flat end plate of the shell, a reinforcing or compensating plate is not so necessary and the shell plate may be simply flanged inwards as shown at (*a*) in Fig. 143.

A circular manhole is shown at (*b*) in Fig. 143. This is a common form in Cornish and Lancashire boilers. As in Fig. 142 the shell S is reinforced by a plate A. The mouthpiece P is of wrought iron or wrought steel. The cover B is pressed to shape from a wrought iron or mild steel plate. The contact surfaces of the flanges of the cover and mouthpiece are machined.

At (*c*), Fig. 143, is shown a mudhole or handhole and cover. The cover B and the bolt C in this example are of wrought iron in one piece.

Manhole and other openings are made steam tight by inserting in the joints asbestos board with interwoven wire gauze.

112. Bases for Boiler Mountings.—The various valves and other boiler mountings are not usually bolted direct to the boiler shell but to suitable bases which are riveted to the shell. Three forms of bases for boiler mountings are shown in Fig 144. At (*a*) is shown a standpipe which is a wrought iron or mild steel forging. The bottom flange is either single or double riveted to the boiler shell and the mounting is bolted to the top flange. At (*b*) is shown a saddle which is pressed to shape from a wrought iron or mild steel plate. At (*c*) is shown a block which is a steel casting, the upper part being provided with slots to receive the heads of T-headed bolts.

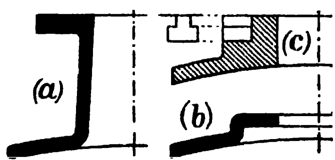


FIG. 144.

113. Boiler Tubes.—The tubes in marine boilers of the Scotch type are made of wrought iron, and with forced draught, which is now general, they have an external diameter of $2\frac{1}{2}$ inches. In length they vary from 7 feet 6 inches to 8 feet 3 inches. About 68 per cent. of all the tubes in a boiler are plain tubes No. 8 Imperial Wire Gauge (0·16 inch) thick. These plain tubes are enlarged $\frac{1}{16}$ inch on the diameter at their front ends as shown in Fig. 145, and they are simply expanded into the tube plates. In Figs. 145, 146, and 147, F is the front tube plate and B the back tube plate.

The stay tubes are screwed into both tube plates. At the front ends the stay tubes are staved up on the outside as shown at F in Figs. 146 and 147. At the back ends the stay tubes may also be staved up on the outside as shown at B in Fig. 146 or they may be staved up on the inside as shown at B in Fig. 147. In any case the diameter over the screw threads at the back end must not exceed the diameter under the threads at the front end. These tubes are screwed

ten threads per inch. Stay tubes of different thicknesses are used in the same boiler, the thinnest being $\frac{1}{4}$ inch and the thickest generally $\frac{3}{8}$ inch. To give additional stiffness to the front tube plate between the nests of tubes a number of the adjacent tubes are frequently stay tubes with nuts as shown at F in Fig. 147, but there are no nuts at the back end.

Locomotive boiler tubes are now seldom less than $1\frac{3}{4}$ inches in external diameter and they are very frequently 2 inches in diameter

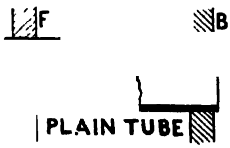


FIG. 145.

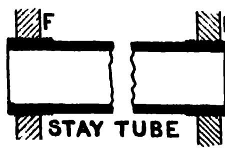


FIG. 146.

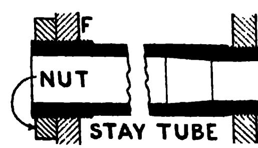


FIG. 147.

and sometimes $2\frac{1}{4}$ inches in diameter. The tubes are pitched zigzag with a water space between them of about $\frac{3}{4}$ inch. They have a thickness of about 1–10th of an inch and their length is from 70 to 100 times their diameter. In British and Continental practice the material of the tubes is commonly brass, but copper, wrought iron, and mild steel tubes are also used to a considerable extent. In American locomotive practice wrought iron tubes are most commonly used but steel tubes are also adopted.

At the smoke-box end the tubes are generally enlarged by $\frac{1}{16}$ inch on the diameter for about 3 inches of their length as in the marine boiler plain tubes shown in Fig. 145. The tubes are expanded into the tube plates at both ends.

Fig. 148 shows a tube swaged down $\frac{1}{8}$ inch on the diameter at the fire-box end, while Fig. 149 shows a tube swaged down and beaded over

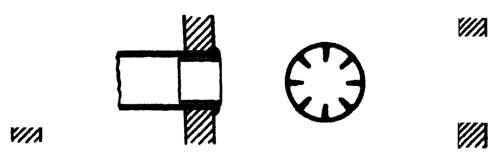


FIG. 148. FIG. 149. FIG. 150. FIG. 151.

at the fire-box end. Brass and copper tubes are beaded over and have steel ferrules driven into them at the fire-box end as shown in Fig. 150.

The *Serve boiler tube* is shown in Fig. 151. This form of tube has a number of internal ribs which extend longitudinally nearly the whole length of the tube and radially about half-way to the centre. These ribs abstract heat from the gases passing through the tube and conduct it to the water outside. Serve tubes, $2\frac{3}{4}$ inches in diameter, have been extensively used in French locomotive boilers and to a very limited extent in British locomotive and marine boilers.

114. Steam Collecting Pipe.—As the steam leaves the water in the boiler it carries with it a certain amount of water, but this water, having a much higher specific gravity than the steam, tends to fall

back as the steam ascends, and the higher the steam rises in the steam space the less water will be suspended in it. The steam should therefore be taken from the boiler as high as possible above the water level. It is most important that the steam should leave the boiler as dry as possible.

Priming, as the carrying away of water in the steam is called, is considerably reduced by placing in the top of the steam space of the boiler a horizontal perforated pipe, closed at its ends, which will collect the steam from different parts of the boiler and lead it to the boiler stop valve. Collecting the steam over a considerable field causes the flow of steam from the water in the boiler to be more uniformly distributed and this promotes the production of dry steam.

An example of a *steam-collecting pipe*, or *anti-priming pipe*, or *dry pipe* is shown in Fig. 152. AB is the collecting pipe, which in this case is made of cast iron. A branch C on AB fits into the standpipe D upon which the stop valve is fixed. The collecting pipe is secured

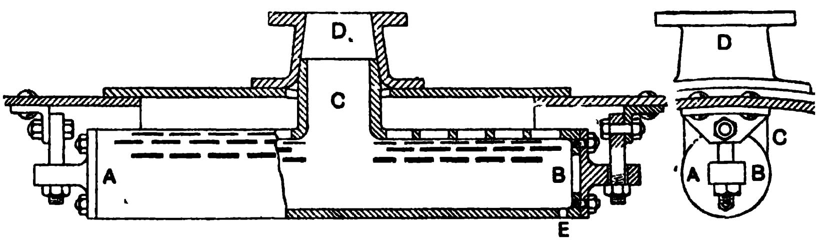


FIG. 152.—Steam collecting pipe.

to the boiler shell by bolts at its ends as shown. In the upper part of AB there is a large number of narrow rectangular holes which admit the steam. Entering the collecting pipe in a downward direction, the steam throws out water suspended in it, and this water returns to the boiler through two small drain holes E in the bottom of the pipe at its ends.

115. Steam Domes.—The only reason for fitting a dome to a steam boiler is to provide a space at a reasonable distance above the water level from which the steam may be taken comparatively free from water. Except in the case of locomotives, domes are now seldom fitted to steam boilers, and a considerable number of locomotives are now made without domes.

The dome is a source of weakness to the shell and adds to the cost of the boiler. A dome, of course, increases the steam space of the boiler, but this is unimportant since the amount of steam which a dome of ordinary size will hold would generally be a very insignificant supply for the engine. A perforated collecting pipe such as that described in the preceding Article makes a dome unnecessary in most boilers.

Two forms of steam dome for locomotive boilers are shown in Figs. 153 and 154. For main line locomotives these domes are generally from 20 inches to 26 inches in diameter, but in large boilers they are sometimes as much as 30 inches in diameter. The dome shell

is generally made of steel but if welded it should be made of wrought iron. The dome cover shown in Fig. 153 may be a steel casting or it

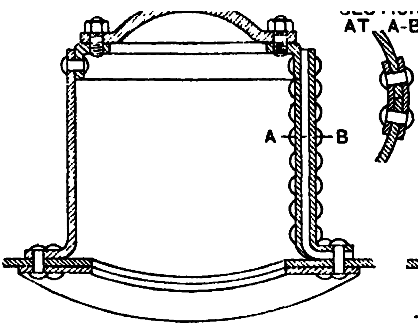


FIG. 153.

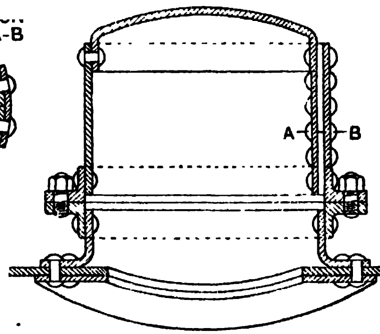


FIG. 154.

may be pressed from a mild steel plate. The bolted joints are faced true and smooth so that no jointing material other than boiled oil or black lead is necessary.

116. Some Locomotive Boiler Details.—Figs. 155 to 158 show various forms of *fire-hole joints* between the inner and outer fire-box plates. In each of these illustrations the inner fire-box plate is on the left-hand side. In the designs shown in Figs. 155 and 156 a welded wrought iron ring is placed between the plates of the inner and outer fireboxes and riveted to them. Fig. 157 shows a joint which was introduced by the late Mr. F. W. Webb of the London and North Western Railway and which is now largely used. An American design is shown in Fig. 158.

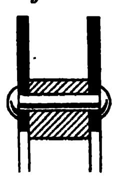


FIG. 155.

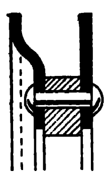


FIG. 156.

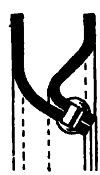


FIG. 157.

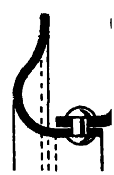


FIG. 158.

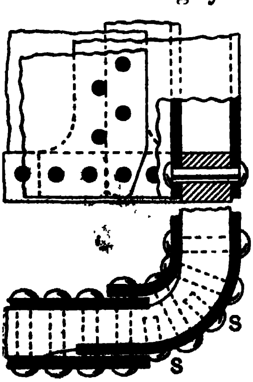


FIG. 159.

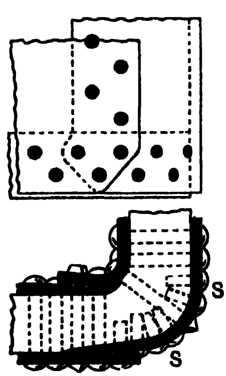


FIG. 160.

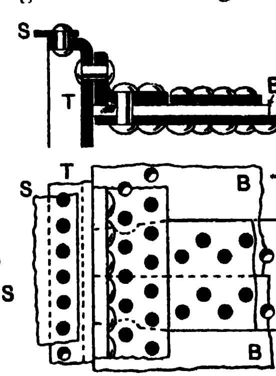


FIG. 161.

Fire-holes, if circular, have a diameter of 14 inches to 16 inches.

If not circular the width is from 15 inches to 18 inches and the height 11 inches to 14 inches.

The inner and outer fire-boxes are riveted together at the bottom with a *foundation ring* between them. This ring is generally a wrought iron or mild steel forging. The riveting may be single or double. Fig. 159 shows the joints at a corner of the foundation ring with single riveting. Another example is shown in Fig. 160. This has double riveting. Where the rivets cannot be taken right through, studs S are screwed in and riveted over.

Three ways of connecting the smoke-box shell to the barrel are shown in Figs. 161, 162, and 163. In these illustrations B is the barrel, T the front tube plate, and S the smoke-box shell. In Fig. 161 the smoke-box shell is riveted to the tube plate which is flanged outwards for that purpose. The tube plate in this case is connected to the barrel by means of an angle ring which is double riveted to the barrel and single riveted to the tube plates. The section in Fig. 161 is taken at the longitudinal joint of the barrel. The smoke-box shell may be riveted directly to the barrel as shown in Fig. 162 or an intermediate ring may be introduced to increase the diameter of the smoke-box as shown in Fig. 163.

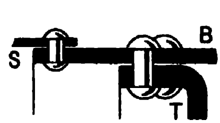


Fig. 162.

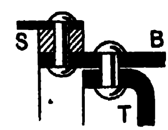


Fig. 163.

CHAPTER VIII

STEAM BOILER MOUNTINGS

117. Water-Level Indicators.—An early form of water-level indicator, now rarely used, consists of a float inside the boiler attached to a thin brass rod which passes vertically through a gland in the boiler shell. The upper end of this rod is attached to a chain which passes over a pulley from which it is led to the front of the boiler where it passes over another pulley from which it hangs; the hanging part of the chain carries a balance weight and, at its lower end, a pointer which, moving over a vertical scale, shows the water level in the boiler. A serious defect of the float gauge is the friction of the rod in the gland which may cause erroneous indications of the water level.

Try cocks, generally three in number, placed at different levels on the front end of the boiler are sometimes used to determine approximately the water level.

The most satisfactory water-level indicator, and the one now almost exclusively used, is the *glass tube water gauge*, by means of which the exact level of the water in the boiler is constantly shown.

A good form of glass tube water gauge, although not of the simplest construction, is shown in Figs. 164 and 165. This is Hopkinson's "absolute" water gauge made by Messrs. J. Hopkinson & Co. of Huddersfield. AB is the front end plate of the boiler and W is the water level, C is a strong glass tube whose ends pass through stuffing boxes in hollow gun-metal castings having flanges D and E for bolting to the boiler. F and H are cocks which control the passages between the boiler and the glass tube. When these passages are open the handles of the cocks F and H are vertical as shown. The water then stands in the glass at the same level as in the boiler. A third cock J, called a blow-through cock, is ordinarily closed, its handle being then vertical as shown.

The complication of the gauge illustrated over the simpler forms of the glass water gauge is due to the arrangement for automatically shutting off the steam and water supply to the glass tube should the latter, from any cause, get broken. The upper and lower castings are connected by the hollow column K. Balls L and M are in the positions shown when the gauge is in normal working condition. Should the glass tube get broken the rush of water in the bottom

passage carries the ball L into the position shown by the dotted circle and shuts off the water. At the same time the steam rushing through the upper passage aided by the water rushing upwards through the column K drives the ball M into the position shown by the dotted circle and shuts off the steam. The attendant may then approach the gauge with perfect safety and shut the cocks F and H and then proceed to renew the glass tube. Screwed plugs N, O, P, and R are convenient for constructional purposes and also give access to the various passages for the purpose of clearing out, when necessary, any sediment which may have lodged in them. The passages may however be kept quite clear by frequent blowing through. In blowing through the cock J is opened and, first F is closed and H opened, then H is closed and F opened.

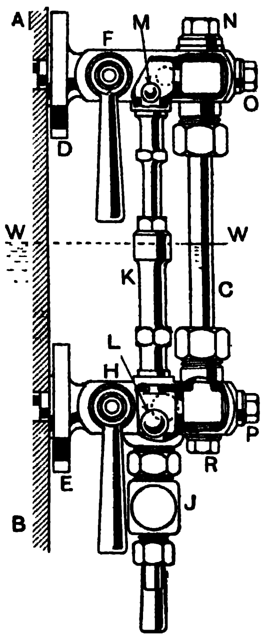


FIG. 164.

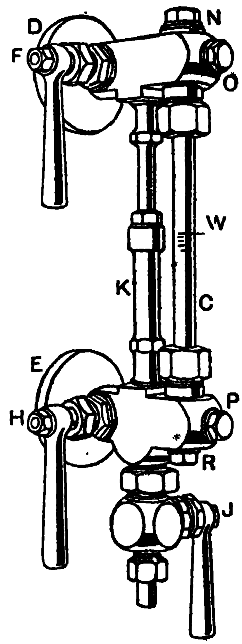


FIG. 165.

The construction of the gauge described above is simplified by dispensing with the ball M, the column K is then no longer required. In the gauge so simplified only the water will be shut off when the glass tube is broken, but there is much less danger from the rush of steam from the top than from a rush of water from the bottom because the water as soon as it escapes into the atmosphere flashes into steam, the volume of which is much greater than the volume of steam issuing from the top of the gauge.

The stuffing boxes holding the ends of the glass tube are generally packed with rubber rings the best form of which is shown in Fig. 166. The rubber ring S has a flange above which there is a brass ring T held down by the gland nut U. The part of the packing ring beyond the flange is conical on the outside but the inside fits the tube against which the ring is forced by the steam or water pressure on the conical surface.

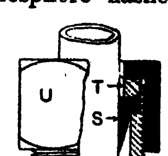


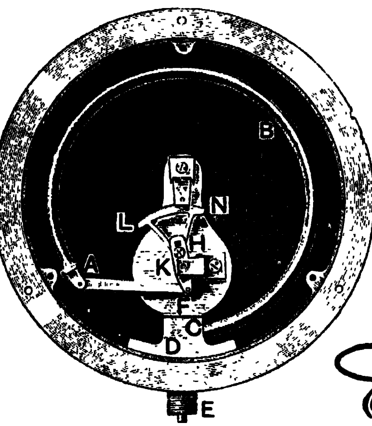
FIG. 166.

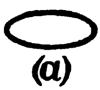
Glass tube water-level gauges for high pressures should be provided with guards made of thick toughened glass which cover the front and

sides of the glass tube. The boiler attendant is then protected from the flying fragments of glass when a glass tube bursts.

118. Pressure Gauges.—The most common form of pressure gauge is that known as the *Bourdon gauge.* This gauge as made by Messrs. Dewrance & Co. of London is shown in Figs. 167 and 168. The interior mechanism is shown in Fig. 167 which is a view looking from the back of the gauge, the back cover being removed. Fig. 168 is a front view showing the dial and pointer.

The essential feature of a Bourdon gauge is the Bourdon spring tube[1] ABC, an enlarged cross section of which is shown at (*a*). As made by Messrs. Dewrance & Co. this tube is of a special quality of bronze and is solid drawn. One end A is closed by a plug while the other end C is firmly secured to a hollow block D which is attached to the casing by the screws S. The gauge is connected by





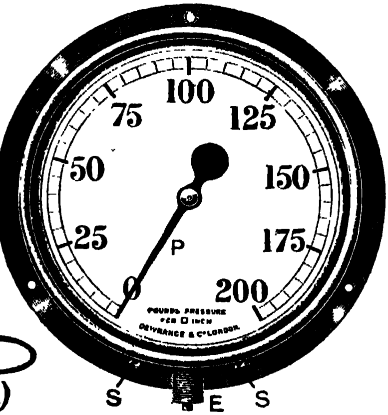


FIG. 167. FIG. 168.

means of the screw E to the top of a siphon to be presently described. This siphon, which is connected to the steam space of the boiler, contains water of condensed steam which enters the gauge at E and fills the Bourdon tube. The pressure of the water in the tube makes its cross section more nearly circular and this change of cross section is accompanied by a flattening of the curve in the other direction. The tube being fixed at C, the end A moves outwards and the movement of A is practically proportional to the difference between the internal and external pressures on the tube. Since the tube is surrounded by the atmospheric air the movement of A is a measure of the pressure in the interior of the tube *above that of the atmosphere.* Hence *absolute pressure* is equal to *gauge pressure* plus *atmospheric pressure.*

The motion of the end A of the Bourdon tube is magnified by the mechanism shown which consists of the connecting rod AF, the lever

[1] Schinz, a German engineer, designed this tube in 1845. In 1850 Bourdon, a Parisian instrument maker, by arrangement with Schinz, patented the tube which has since then been known as the Bourdon tube.

FH pivoted at K and carrying the toothed sector HLN which gears with a small pinion fixed to the spindle which carries the pointer P, the position of which on the dial indicates the pressure in pounds per square inch.

The pressure gauge is generally constructed to indicate up to double the maximum working pressure so that the pointer is vertical when the safety valve is about to blow off. The dial in Fig. 168 would be for a working pressure of 100 pounds per square inch and the number 100 at the top of the dial should be in red, the other numbers being in black.

As already stated the pressure gauge is connected to the boiler through a siphon, a good form of which is shown in Fig. 169. This is a gun-metal casting having a flange M for bolting to the end of the boiler shell at a point well above the water level W in the boiler. The gauge is connected to the siphon at O. A plug R closes a short branch to which an inspector's standard gauge may be periodically attached for the purpose of testing the accuracy of the boiler pressure gauge. A plug T closes an opening at the bottom of the siphon which gives access to it for the purpose of clearing it, when necessary, of any sediment which may accumulate in it. The passages between the boiler and the boiler gauge and the inspector's gauge are controlled by the three-way cock V.

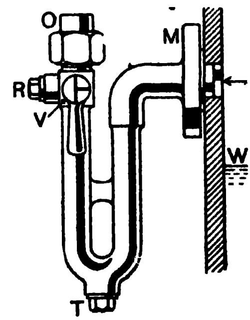


Fig. 169.

Since the water in the siphon prevents the hot steam from entering the Bourdon tube the gauge is kept comparatively cool. Continuous overheating of the Bourdon tube would injure it and permanently affect the accuracy of the gauge. When in use the siphon should be cool enough to be grasped by the hand without discomfort.

119. Safety Valves.—All except small boilers should have at least two safety valves which will open when the boiler pressure exceeds the working pressure. These valves are almost exclusively of the disc type with narrow flat or conical seats, and they should be large enough to discharge the whole of the steam as rapidly as it is generated when the boiler is working at full power.

Safety valves must be placed on mountings directly on the boiler.

A disc valve is full open when its lift is one-fourth of its diameter if the passage under it is not obstructed by wings or other arrangements for guiding the valve. When there are such obstructions the full open lift is less than one-fourth of the diameter. In ordinary working the lift of a safety valve is generally less than one-eighth of an inch. By the diameter of a disc valve is meant the internal diameter of the seating of the valve.

Safety valves are loaded either by weights or springs acting either directly on the valves or through levers.

As the valve rises the load is lifted and in the case of a spring load the force exerted by the spring increases as the valve lifts. This

increase of force exerted by the spring retards the lifting of the valve after it has opened. To overcome this difficulty, valve discs are frequently enlarged above their faces as shown at (*a*) and (*b*) in Fig. 170. The reaction of escaping steam on the enlargement or lip of the disc further assists in the lifting of the valve and thus retards accumulation of pressure. A similar effect is produced in the inverted cup-shaped valve shown at (*c*).

The lip on a valve is made more effective by adding to the seating a guide to direct the escaping steam on to the lip as shown at (*a*). A still more effective design for this purpose is shown at (*d*) where the lip is on a separate piece which is adjustable as to height. Valves of this design are known as "pop valves"; they give a large lift very rapidly as soon as they begin to open.

When the valve seating is a separate piece it should be firmly secured to the casing to prevent it being lifted with the valve by the steam pressure. For smaller valves the seating may be screwed into

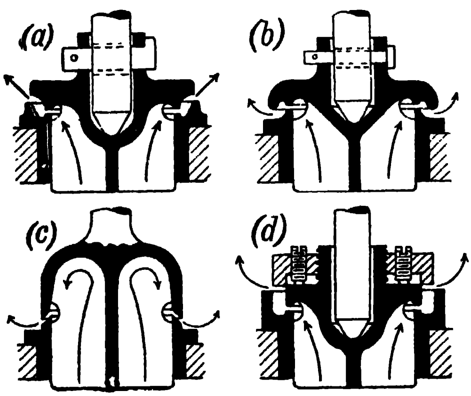


FIG. 170.

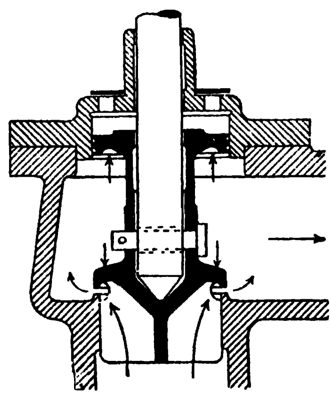


FIG. 171.

the casing as in Fig. 172. For larger sizes the seating may be secured by studs passing through lugs on the flange at the top of the seating.

When the steam escaping through a safety valve is led through a long pipe into the atmosphere outside the boiler room there is a considerable back pressure on the valve after it opens and the lip on the valve is less effective. This has led Mr. J. Hamilton Gibson to design the valve shown in Fig. 171.[1] There is no lip on this valve but the valve extends upwards as a sleeve into a balance piston which is a loose fit, without packing, in a cylinder formed in the cover of the valve chest. Any steam leaking past the piston escapes through the holes shown. These holes are covered by a thin loose plate which excludes dirt. In the design shown in Fig. 171, the valve chest and its cover as well as the valve and piston are made of gun metal.

Since steam escaping through the safety valves is wasted, it is one of the tests of good stoking that these valves should seldom operate through excess of pressure. But to ensure that the safety valves are

[1] See *Engineering*, Feb. 26, March 12, and March 19, 1909.

in working order they should be frequently tested by lifting the levers or operating the easing gears provided.

120. Lever Safety Valves.—A good design of a lever safety valve loaded with a weight is shown in Fig. 172. The valve V and its seat S are of gun-metal. The casing B is generally made of cast iron or cast steel. The lever, made of wrought iron or mild steel, has its fulcrum at F, and is loaded by the weight W. The thrust is transmitted to the valve through the short hinged strut R. The pins of the pin joints are made of gun-metal to prevent corrosion. But these pins may be of steel if the holes in the lever are bushed with gun-metal. The pins must be an easy fit in the lever.

The load W is secured to the lever by a pin and padlock as shown. G is a fork which acts as a lateral guide to the lever. Frequently this fork is bridged over the lever at the top for the purpose of preventing the valve being blown away in the event of the lever getting broken or the weight removed. This is a very objectionable arrangement because it provides a very easy means of unauthorized overloading of

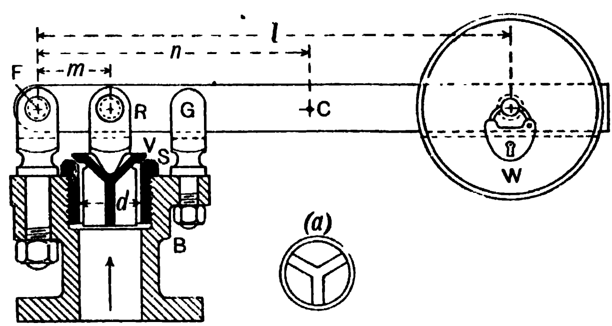


FIG. 172.—Lever safety valve.

the valve by placing a piece of wood or metal on the lever under the bridge and thus locking the valve on its seat. With a wrought iron or steel lever and the weight firmly secured to it the need of any guard to limit the lift of the valve is small.

A plan of the valve, looking upwards, is shown at (*a*).

Let d = diameter of valve in inches.
W = load on outer end of lever in lb.
w = weight of lever in lb. acting at its centre of gravity C.
w_1 = combined weight of valve and strut R and its pin in lb.
p = pressure of steam, in lb per sq. in. when the valve is about to open.

Then, taking the dimensions l, m, and n (Fig. 172) in inches, and taking moments about F,

$$Wl + wn + w_1m = \frac{\pi}{4}d^2pm$$

121. Dead-Weight Safety Valves.—Fig. 173 illustrates a *dead-weight safety valve* made by Messrs. J. Hopkinson & Co., Huddersfield.

A is the valve which rests on the valve seat B which is secured to the top of the vertical pipe C. This vertical pipe has a flange at the bottom for bolting to a seating block on the top of the boiler shell. The particular valve shown has an enclosed discharge which leads the escaping steam to a pipe, connected to the discharge casing at H, which carries the steam to a convenient point outside the boiler house.

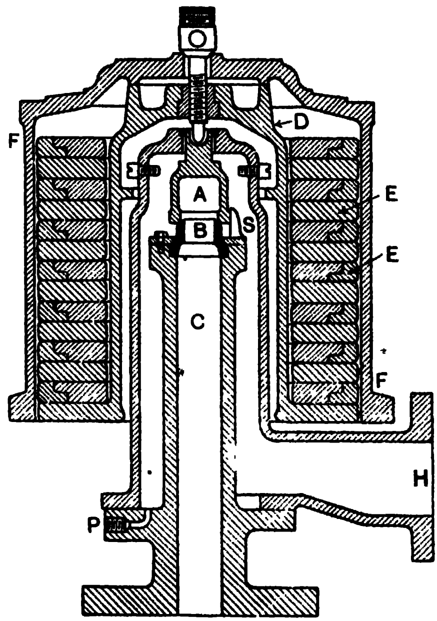


Fig. 173.—Dead-weight safety valve.

The valve seat is fastened down by a ring and screws as shown. The securing ring has feathers S cast on to it which act as guides for the valve A. The valve is also guided by a ring in the top of the discharge casing.

Suspended from the top of the valve is the weight carrier D. The weights E are cast iron rings which are enclosed in the cast iron case or cover F. The load on the valve is made up of the weight of the carrier, the rings, and the cover, and the weight of the valve itself, and this load balances the total steam pressure on the valve when it is blowing off.

The valve and the weights which it carries are prevented from being blown away by a stop ring cast on the inside of the weight carrier D and screws in the discharge casing as shown. This precaution is necessary in the event of a rush of very wet steam due to serious priming. The discharge casing is drained by a pipe connected at P.

The centre of gravity of the dead weight is well below the point of suspension, and this ensures that the weight hangs vertically.

Dead-weight safety valves are only suitable for stationary boilers. They have the great advantage that they cannot be readily tampered with, since any added weight must be equal to the total increased pressure of the steam on the valve. For example, a valve 3 inches in diameter has an area of about 7 square inches, and to increase the blowing off pressure by, say, 5 lb. per square inch, would require an addition of $5 \times 7 = 35$ lb. to the weight, an addition which would be readily observed.

The objection to dead-weight safety valves is the great weight which has to be carried. For a working pressure of 150 lb. per square inch a 2-inch valve would require a total weight of 471 lb., while a 3-inch valve would require a load of 1060 lb.

122. Combined High-Steam and Low-Water Safety Valves.—Boilers of the Cornish and Lancashire types are generally provided with a safety valve which allows the steam to escape when the water level becomes too low and so prevents the collapse of the furnace tubes should they become overheated due to shortness of water. This low-water safety valve is usually combined with another safety valve which blows off when the steam pressure exceeds the working pressure.

The best known combination of these valves is Hopkinson's which

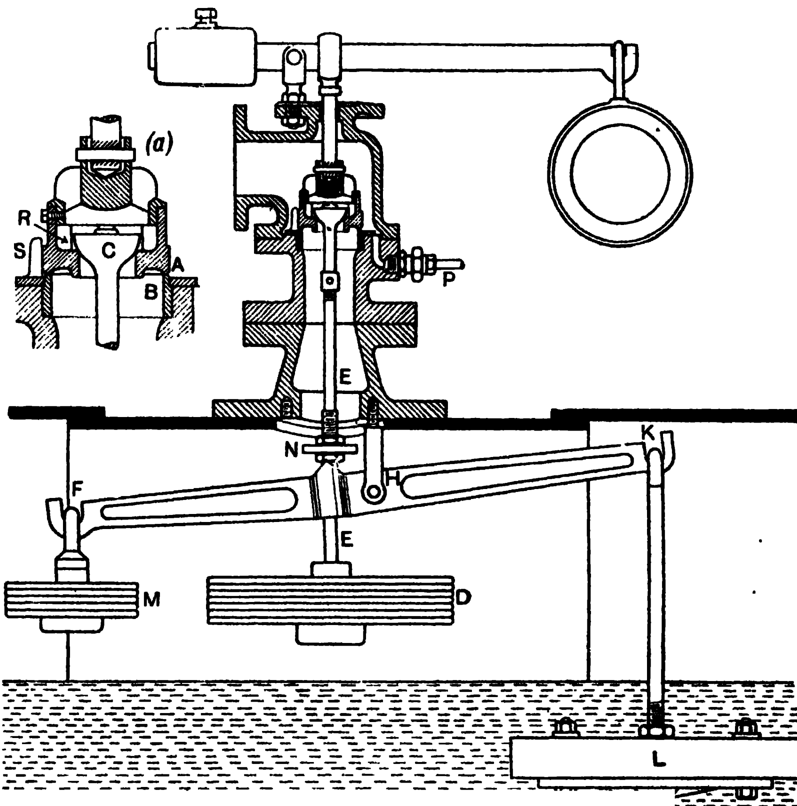


Fig. 174.—Combined high-steam and low-water safety valve.

is shown in Fig. 174. A section of the valves themselves is shown to a larger scale at (a). A is a valve resting on the seat B. The edge of a central opening in A forms the seat for a hemispherical valve C which is loaded directly by the weight D attached to the lower end of the rod E which is a continuation of the spindle of C. Inside the boiler there is a lever FK whose fulcrum is at H. A large earthenware float L is suspended from the end K of the lever FK, and when this float is fully immersed in the water it is balanced by the weight M

suspended from the other end of the lever. When the water level falls and the float L is sufficiently uncovered the weight M will not be sufficient to balance L, and L will descend. On the lever to the left of the fulcrum there are two projections, one on the front and the other on the back of a boss on the lever through which the rod E passes. The descent of the float L causes these projections on the lever to come in contact with a collar N attached to the rod E, and the valve C is lifted and steam escapes.

When the projections on the lever FK are clear of the collar N the valve A acts as an ordinary safety valve loaded partly by the weight D and partly by the loaded lever above the valve casing. P is a drain pipe to carry off any water which may be deposited in the valve casing. Ribs R on the inside of the valve A act as guides for the valve C, and feathers S cast on the ring which secures the valve seat B form guides for the valve A.

123. Spring-Loaded Safety Valves.—For all but stationary boilers springs are necessary for loading the safety valves. It has already been mentioned (Art. 119) that an objection to a spring load is that as the valve opens the force exerted by the spring increases. This defect may, however, be made comparatively unimportant by using a long spring. For the same increase of load on a spring the alteration in its length is directly proportional to the length of the spring.

The springs used for loading safety valves are nearly always helical springs made of round or square spring steel rod. The springs as a whole may be either in tension or compression, but most frequently they are placed in compression.

The type of safety valve introduced by the late Mr. John Ramsbottom on the London and North Western Railway is the one most generally used on British locomotives. The form of Ramsbottom safety valve used on the Great Eastern Railway is shown in Fig. 175. There are really two valves V, of the same size, having their seatings in the upper ends of two hollow standards C which are united by the bridge B and a flange or base A, the base A being bolted to a shallow pressed steel stool on the top of the boiler over the fire-box. The valves are held down by the helical spring S and lever L. The lever has two pivots P and Q of which Q is forged on the lever and P is jointed to it. These pivots bear on recesses in the valves. The upper end of the spring is hooked to an arm R of the lever midway between the valves while the lower end is hooked to the shackle D which is secured to the bridge B.

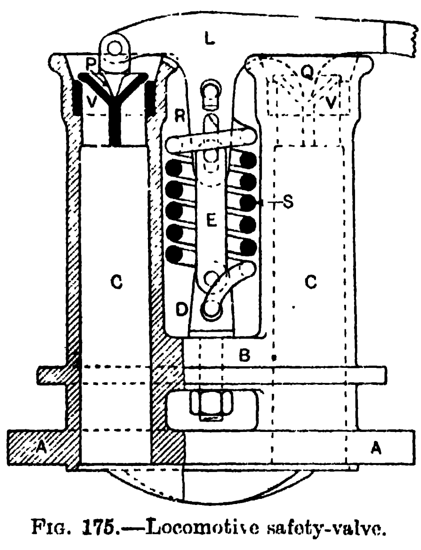


FIG. 175.—Locomotive safety-valve.

To prevent the valves from being blown away in the event of the spring breaking there are two links E, one behind the other on either side of the lever, connected by pins at their ends. The lower pin passes through the shackle D while the upper pin passes through a slot in the arm R of the lever. The lever L has an extension to the right which projects into the cab where it may be lifted or pulled down in order to test whether the valves are free to act properly.

The "*Pop*" *safety valve* as made by the Lunkenheimer Company of Cincinnati, U.S.A., is shown in Fig. 176.[1] A is the body and B the cover. C is the seating and D the valve, the bearing surface between them being inclined at 45°. The area presented to the escaping steam by the lip E of the valve increases the lift. The action of the lip can be adjusted without opening up the valve, as follows. Remove the set-screw F, pass a pointed rod through the aperture and using it as a ratchet work the notched screwed ring H round, thus raising or lowering it, and decreasing or increasing the amount of steam diverted from the lip through the lateral openings in the prolongation of the seating which directs the steam on to the lip. The ring H is screwed upwards if the pressure is not sufficiently relieved and downwards if it is relieved too much. After adjusting the ring H replace the set-screw F so that the point catches in one of the notches of the ring.

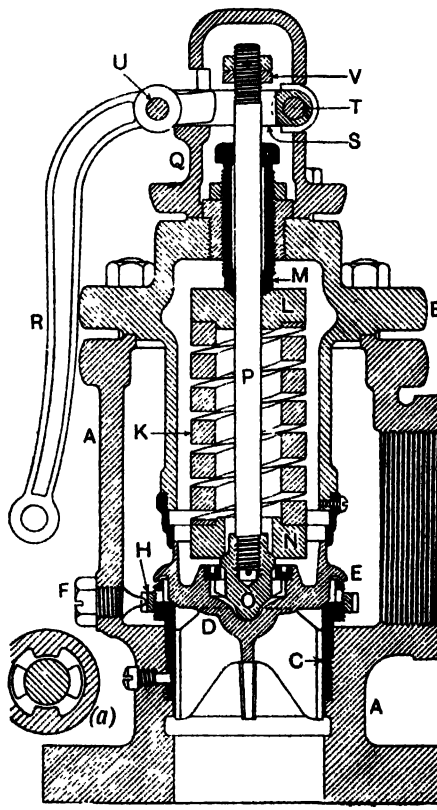


FIG. 176.—"Pop" safety valve.

The spring K presses on washers L and N of which L has a spherical bearing on the adjusting screw M while N has a spherical bearing on the piece O which presses on the valve. The lifting stem P is screwed into the piece O and locked with a rivet passing transversely through both. The piece O has lugs on it which lie in recesses in the valve as shown at (*a*). This arrangement enables the valve to be rotated on its seat by turning the stem P after removing the cap Q.

[1] Prepared from a working drawing kindly supplied by the Lunken Company, London.

The valve may be lifted from its seat by means of the lever R and forked lever S. The lever S is jointed to the cap Q by the pin T and to the lever R by the pin U. Turning the lever R outwards causes the lever S to press upwards on the nut V which is locked on the lifting stem P.

It will be seen that the spring K is completely protected from contact with the escaping steam.

By locking the pin T with a special nut or with a padlock the load on the valve cannot be tampered with.

Boilers in the mercantile marine are usually fitted with spring-loaded safety valves of the enclosed lock-up type shown in Fig. 177,

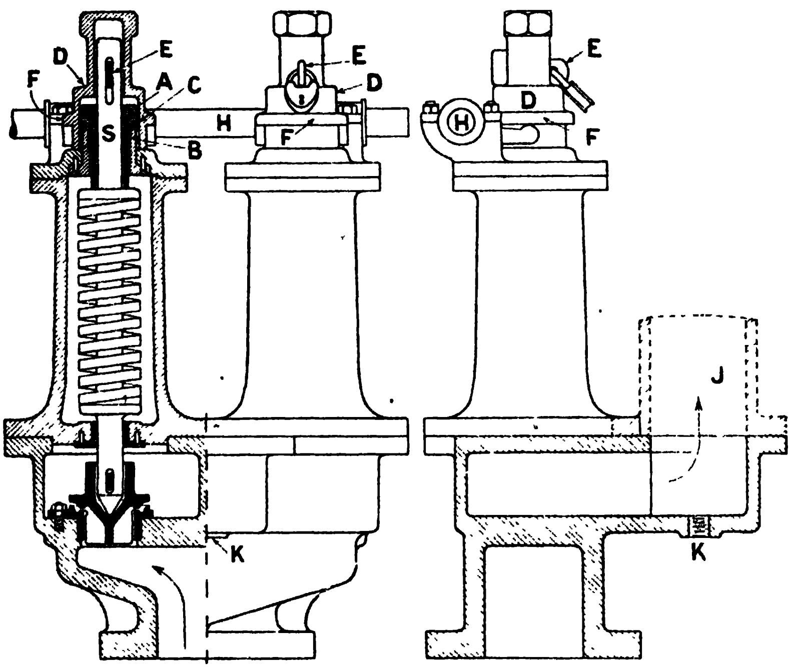


FIG. 177.—Marine safety valve.

which illustrates the *marine safety valve* made by Messrs. Alex. Turnbull & Co., Glasgow. The force exerted by the spring is adjusted by the screw A which has a hexagonal head and works in the nut B. When finally adjusted the screw A is removed and a ring C of the correct thickness is placed on top of the nut B ; the screw A is then reinserted and tightened up against the ring as shown,

The adjusting screw and the top of the valve spindle S are enclosed by the cap D. A gib-headed cotter E passes through the cap and spindle, there being considerable clearance in the cotter hole in the spindle so that the valve and spindle may rise under the action of the steam pressure. The cotter, which is secured with a padlock, prevents the removal of the cap D and therefore prevents access to the adjusting

screw A. The cap has a hexagonal head to receive a spanner which may be used to rotate the cap, and because of the cotter this rotation will be communicated to the spindle and valve.

On the cap D there is a collar F under which are the prongs of a fork rigidly attached to the shaft H which may be twisted with suitable gear from the stokehold floor. The twisting of the shaft H in the proper direction moves the fork, lifts the cap D and opens the valve.

Fig. 177 shows two valves mounted on the same base. For large boilers there may be three or even four valves grouped together on the same base. The steam escaping through the valves is led away by the waste pipe J. Water accumulating in the valve chest is carried away by a drain pipe attached at K.

124. Junction and Stop Valves.—A valve placed directly on a boiler and connected to the steam pipe which leads to the engine is called a *junction valve*. Where there are a number of boilers delivering steam into one main steam pipe each boiler must have its own junction valve.

A valve placed in the steam pipe leading to the engine and generally placed near to the engine is called a *stop valve*, but junction valves are also very frequently called stop valves.

Junction and stop valves are operated by hand and their function is to regulate the amount of steam passing through the steam pipe and to shut it off altogether when required. There is no essential difference between the design of a junction valve and that of a stop valve.

The *Crosby* junction or stop valve is shown in Fig. 178. If used as a junction valve the flange A of the body is bolted to the boiler at the highest part of the steam space. The seat B is screwed into the body by aid of lugs C cast on its interior. The disc D has a renewable disc seat E screwed on to it. The construction of the seats is shown to a larger scale at (*a*). It will be seen that the bearing surfaces of the seats are of double conical form and that there are grooves between the conical surfaces which give a certain amount of spring to the seats. For that reason these valves are called "spring-seated valves."

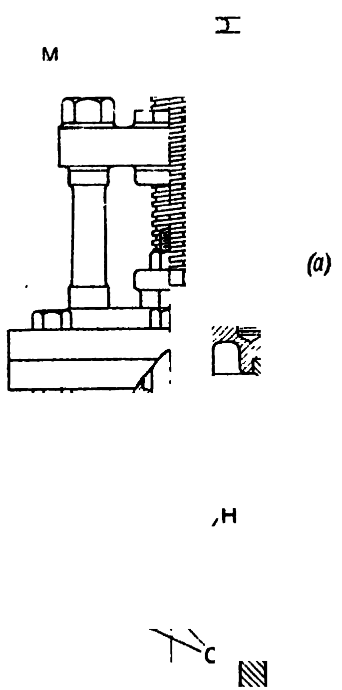


FIG. 178.

The valve disc D is connected to the spindle F by the nut H the lower edge of which comes in contact with a collar on the lower end of the spindle. The spindle as it is raised or lowered carries the disc D with it but is free to rotate within the disc. A square projection K on the under side of the disc enables the disc to be held when the nut H is being screwed in or out.

The spindle passes through a gland and stuffing box in the cover of the body. The upper portion of the spindle is screwed and passes through a nut in a crosshead or yoke carried by two pillars which are screwed into the cover of the body as shown. The spindle is rotated by means of the hand wheel M. By raising the spindle to its full height the upper conical face of the collar L is made to bear on a corresponding seat on the bottom of the stuffing box and the latter may then be repacked under pressure.

In an ordinary single-seated valve the valve, opening gradually, is at first open to a very small extent, the steam rushes through this small opening with a much higher velocity than when the valve is more fully open and equilibrium of pressure is established on both sides; this wire-drawn high velocity steam exerts a cutting action on the bearing surfaces of the valve and its seat.

A design of valve which guards against the above mentioned cutting action of the steam and which has other advantages over the ordinary design of junction or stop valve is shown in Fig. 179. This design is due to Messrs. Hopkinson & Co., of Huddersfield. There are two valve discs A and B with corresponding seats as shown. A is attached to the central spindle C while B is attached to the lower end of the sleeve G whose upper end is attached to the crosshead D. The upper part of the central spindle passes through a screwed sleeve S which is secured to the hand wheel W, but the screw on the lower part of this sleeve has a larger pitch than the screw above it. Of these two screws the upper one works in the crosshead E while the lower one works in the crosshead F.

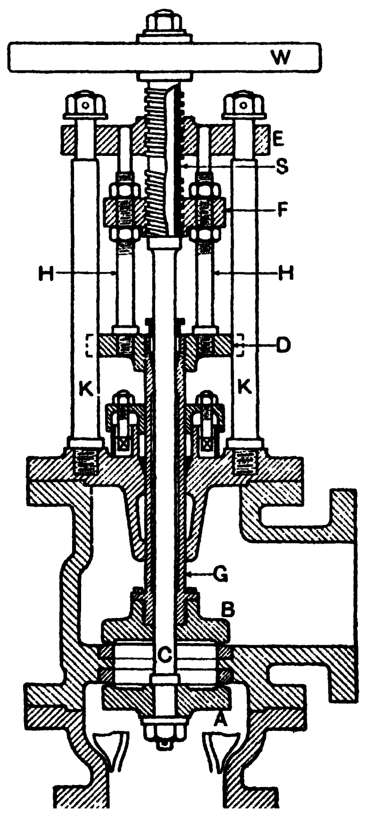


Fig. 179.

The crosshead F is connected to the crosshead D by the rods H. The crosshead E is guided by the pillars K but the upward motion of E is limited by the nuts on the upper ends of the pillars.

As shown in Fig. 179 the valve is closed. If, viewed from above, the hand wheel is turned anti-clockwise the crossheads E and F will rise because the screwed sleeve S cannot descend on account of the steam pressure on the valve disc A. When the crosshead E has reached its highest position, the valve B has opened considerably and any further motion of the handwheel will cause the sleeve S and disc A to descend; at the same time the disc B rises further because the pitch of the screw working in F is greater than the pitch of the screw working in E. It will therefore be seen that no steam passes the edge of B until it is open to a considerable extent and the wire drawing action occurs almost entirely at the edge of A. The disc A therefore protects

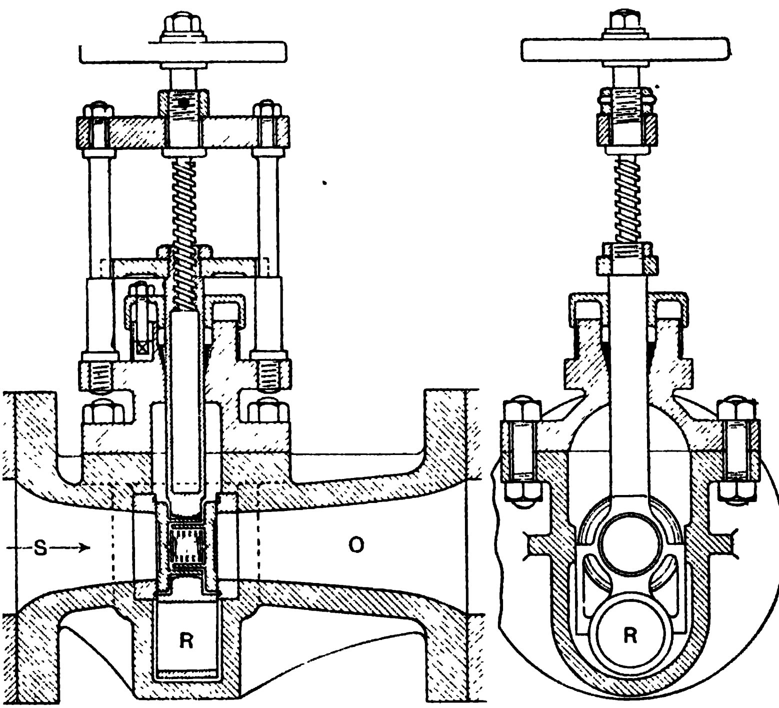


FIG. 180.—Hopkinson-Ferranti stop valve.

the edge of B and its seat from the cutting action of the high velocity wire-drawn steam.

When the valve is closed it is evident that the forces holding the discs A and B on their seats balance one another and there is no stress in the valve casing due to the tightening of the valve discs on to their seats, and troubles due to unequal expansion or contraction are avoided. This design of valve is suitable for large sizes and is made up to 12 inches in diameter.

The *Hopkinson-Ferranti stop valve* is shown in Fig. 180. This is a "straight through" valve having the special feature that the throat where the valve is placed is considerably less in diameter than the pipe

in which the valve casing is placed. Generally the diameter of the throat is half that of the pipe. The inlet passage S to the throat is converging and the outlet passage O is diverging. As the steam passes through the converging passage its velocity and its kinetic energy increase while its pressure energy and pressure decrease. After passing through the throat the velocity and kinetic energy of the steam diminish while the pressure energy and pressure rise, and if the contours of the passages are properly designed and their surfaces are smooth the velocity and pressure of the steam will be only slightly affected by its passage through the valve.

The construction of the valve itself and the method of operating it will be readily understood from the views shown in Fig. 180, but it may be mentioned that the valve carries, suspended below it, a ring R which encircles the throat when the valve is full open and then makes the passage through the throat smooth and continuous.

A Hopkinson-Ferranti valve being much smaller than an ordinary valve for the same size of steam pipe it is lighter and easier to operate and it presents a smaller periphery of bearing for the leakage of steam.

125. Feed Check Valves.—At the boiler end of the delivery pipe from the feed-water pump a non-return valve must be placed as near to the boiler as possible. This non-return valve or *check valve* is usually provided with an arrangement for regulating by hand the extent of its opening because it is important that the water level in the boiler should be maintained as nearly as possible constant.

A good design of feed check valve by Messrs. Hopkinson of Huddersfield is shown in Fig. 181. C is the check valve the lift of which is controlled by an extension of the spindle of the screw down valve V above it. It is most important that the check valve should be kept in perfect condition. To ensure this it may, in the design illustrated, be examined and cleaned or re-ground when the boiler is under steam by closing the valve V, shutting off the feed water, and then uncoupling the elbow E which contains the check valve and its seat. Before doing this, however, sufficient water is fed into the boiler to keep it going while the check valve is disconnected.

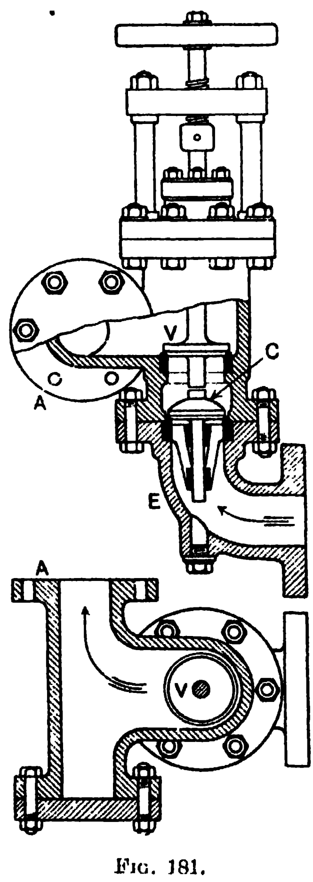


FIG. 181.

The flange A is bolted to the end of the boiler shell at a point from

which an internal perforated pipe leads the feed water and distributes it near the working level of the water in the boiler.

126. Blow-Off Valves.—Periodically it is necessary to empty the boiler in order that it may be cleaned and inspected internally. It is also a common practice periodically to discharge a portion of the water from the bottom of the boiler in order that any sediment which may have been deposited may be carried away. For these purposes a blow-off valve or cock is fitted to the lowest part of the boiler. This valve is either fitted directly to the boiler shell or to a short branch or to an elbow pipe of cast steel as shown at W in the illustration of a Lancashire boiler, Fig. 89, p. 124.

When several boilers are arranged to discharge into the same waste pipe the blow-off valve of each boiler should have connected to it an isolating valve which will prevent water which is being discharged from one boiler being blown into another which may be open for

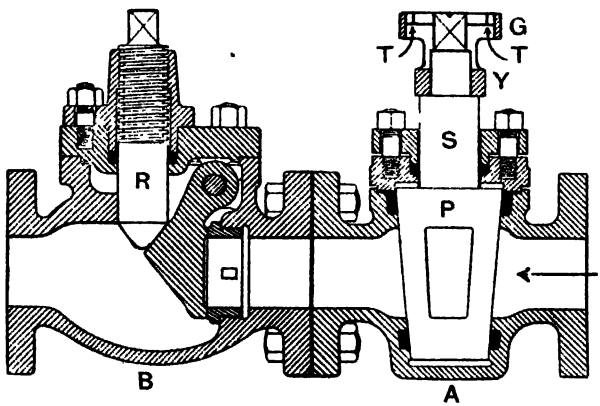


FIG. 182.—Isolating valve and blow-off cock.

inspection, the blow-off valve of which has been inadvertently left open.

Fig. 182 shows a blow-off cock A and an isolating valve B made by Messrs. Dewrance & Co. of London. The cock and isolating valve are shown closed. The plug P of the cock is conical and fits into the body or casing which is packed with asbestos packing in grooves round the top and bottom of the plug as shown. There are also grooves connecting those at the top and bottom which are likewise packed. The asbestos packing is rammed in tight and the plug bears on the packing. Cocks packed in this way keep tight better under high pressure and are more easily operated than unpacked cocks. The shank S of the plug P passes through a gland and stuffing box in the cover. The plug is held down by a yoke Y and two stud bolts, not shown, one behind and the other in front of the shank S. The yoke Y has formed on it a guard G on the inside of which are two vertical slots T through which pass projections on the box spanner used for operating the cock. The use of this guard is to prevent the spanner being removed while the cock is open.

The isolating valve B is of the hinged non-return type and opens outwards, the amount of opening being regulated by the screwed stop R.

When the blow-off cock is not associated with an isolating valve, and when the cock is only partially open, the flow of gritty water scores the plug and the asbestos packing becomes injured with the result that the cock afterwards leaks. With an isolating valve, however, the cock may be opened full while the isolating valve is closed, and the latter is then allowed to open to any desired extent. Again in closing, the isolating valve is first shut and then the blow-off cock is closed. The scoring action of the gritty water on the partially open isolating valve is not a serious matter because it is not so important that this valve should be absolutely water tight.

Fig. 183 shows a blow-off valve used on locomotives on the Great Eastern Railway.

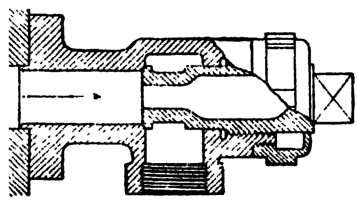


FIG. 183.

127. Fusible Plugs.—The crown of a furnace or combustion chamber should be fitted with a plug held in position by a fusible metal or alloy. This plug under normal conditions is covered with the water in the boiler which keeps the temperature of the fusible metal below its melting point. But should the water in the boiler fall below the "low-water" level the fusible metal is melted by the heat of the furnace, the plug drops out and steam rushes into the furnace and puts out the fire or gives warning to the stoker that the furnace or combustion chamber crown is in danger of being overheated.

Figs. 184 and 185 illustrate two forms of fusible plug patented by the National Boiler Insurance Co., Manchester. A is a hollow gun-

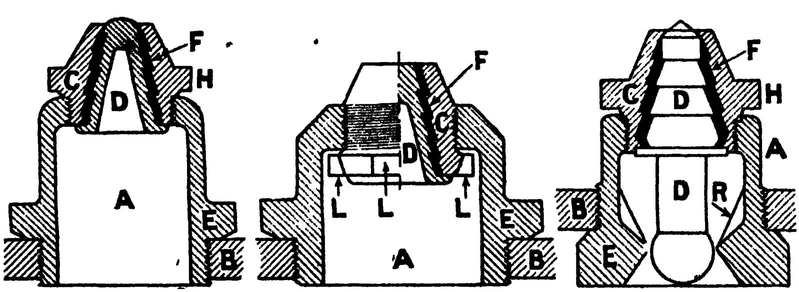


FIG. 184. FIG. 185. FIG. 186.

metal plug screwed into B the furnace crown. C is a second hollow gun-metal plug screwed into the first. D is a third hollow gun-metal plug separated from C by an annulus of fusible metal F. The inner surface of C and the outer surface of D are grooved as shown, so that when the fusible metal is poured in the plugs C and D are locked together. E is a hexagonal flange on A to take a spanner for fixing or removing A. H (Fig. 184) is a hexagonal flange on C for fixing or

removing C. The spaces between the lugs L (Fig. 185) serve to receive the prongs of a special key or spanner when it is required to screw or unscrew the plug C.

To renew the fusible plug only the part C need be taken out. In the design shown in Fig. 185 C may be removed and replaced by operating from the inside of the furnace, but in the design shown in Fig. 184 C can only be operated from the inside of the boiler.

It will be observed that the fusible metal is protected from the fire by the flange on the lower end of D. Also, in the design shown in Fig. 184 there is contact at the top between C and D so that the fusible metal is completely enclosed.

Fig. 186 shows one of the designs of fusible plug patented by the Vulcan Boiler Insurance Co., Manchester. The inner plug D in this design is made of copper and has a prolongation with a spherical end which practically closes the entrance, at the fire side, to the cavity beneath the plug. When the alloy is fused the plug D is not blown right out into the fire but after dropping a certain distance is held suspended, while at the same time a free way is provided for the escaping steam. This is effected by means of the flange on the plug D and the ribs R on the interior of A. The hanging plug is then visible to the stoker when he is firing and warning is therefore given even, when the fire has just been lighted, before sufficient water has been fed into the boiler to cover the furnace crown, in which case the crown might become over-heated before any steam was provided.

The fusible metal is subject to deterioration due to exposure to heat and it should be renewed at intervals of, say, two years.

CHAPTER IX

STEAM BOILER ACCESSORIES

128. Feed Pumps.—The appliances in common use for delivering the feed-water into steam boilers are: reciprocating pumps, rotary pumps, and injectors. The reciprocating pumps are either single or double acting. A single acting or plunger pump delivers water during alternate strokes while a double acting pump delivers water during each stroke.

There are many designs of reciprocating pumps, the difference in design being largely due to the way in which the pump plunger or piston is driven. The pump may be driven through levers or cranks and connecting rods from the main engine, but more generally the pump has its own steam cylinder and is worked independently of the main engine; the pump is then called an *independent feed pump*.

The most common form of independent reciprocating feed pump is that in which the steam cylinder is directly opposite to the water cylinder, the rod of the steam piston being connected directly to the plunger or to the rod of the water piston. An example of this type of pump is illustrated and described in the next Art.

Rotary pumps are generally of the high speed centrifugal type, driven directly by a small steam turbine or by an electric motor.

129. Weir's Feed Pump.—A well known form of direct acting reciprocating feed pump is shown in Fig. 187. The principal feature of this pump is the ingenious valve gear for distributing the steam to the steam cylinder. Before the valve gear in detail is considered, a general description, in the form of a list of parts, will be given as follows:

1. Steam cylinder.
2. Steam piston.
3. Piston rod.
4. Pump cylinder.
5. Pump piston.
6. Pump rod.

7 and 8. Seats for pump suction and delivery valves.

9. and 10. Guards to limit lift of suction and delivery valves.

11. Columns fixing distance between valve seats and valve guards.
12. Screw passing through pump valve chest cover to secure valve seats and valve guards.

13. Main crosshead, a steel forging, screwed the full depth, split on one side and bolted. The ends of the piston rod and pump rod are screwed into this crosshead, butt hard on each other and are locked by the tapered pin *p* which bears on the ends of the rods only.
14. Steam valve chest.
15. Steam stop valve.
16. Exhaust stop valve.
17. Lubricator for steam valves and piston.
18. Valve gear levers.
19. Fulcrum of valve gear levers.
20. Crosshead pin having sliding connection with valve gear levers.
21. Auxiliary steam valve spindle.
22. Bottom spindle.
23. Double joint connecting 21 and 22.
24. Crosshead between levers (18). The bottom spindle (22) passes through this crosshead and the auxiliary steam valve is operated by the crosshead striking the head *h* or nut *n* on the bottom spindle.

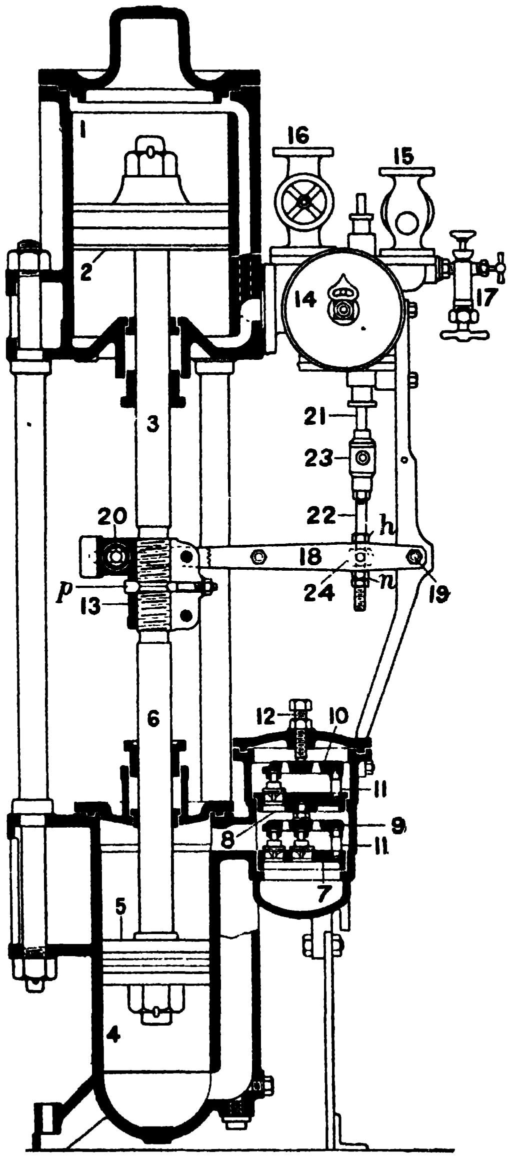


Fig. 187.—Weir's feed pump.

Coming now to the valve gear of the steam cylinder. The valves and valve chest are shown in detail in Fig. 188.

There are two slide valves, the main valve A and the auxiliary valve B. The main valve is cylindrical, the middle part being cut away to give a flat face for the auxiliary valve to work on. The main valve has a shuttle motion and is sometimes called the *shuttle valve.*

In Fig. 188 (*a*) is a cross-sectional elevation of the valve chest and valves; (*b*) is a longitudinal-sectional elevation of the valve chest, the valves being removed in order to show the ports leading to the cylinder; (*c*) is a sectional plan of the valve chest and valves; (*d*) is an elevation of the main valve looking on the flat back upon which the auxiliary valve works; (*e*) is an elevation of the main valve looking on the curved face; (*f*) is an isometric projection of the auxiliary valve.

The main valve A distributes steam to the cylinder. The auxiliary valve B has two functions; it distributes steam to work the main

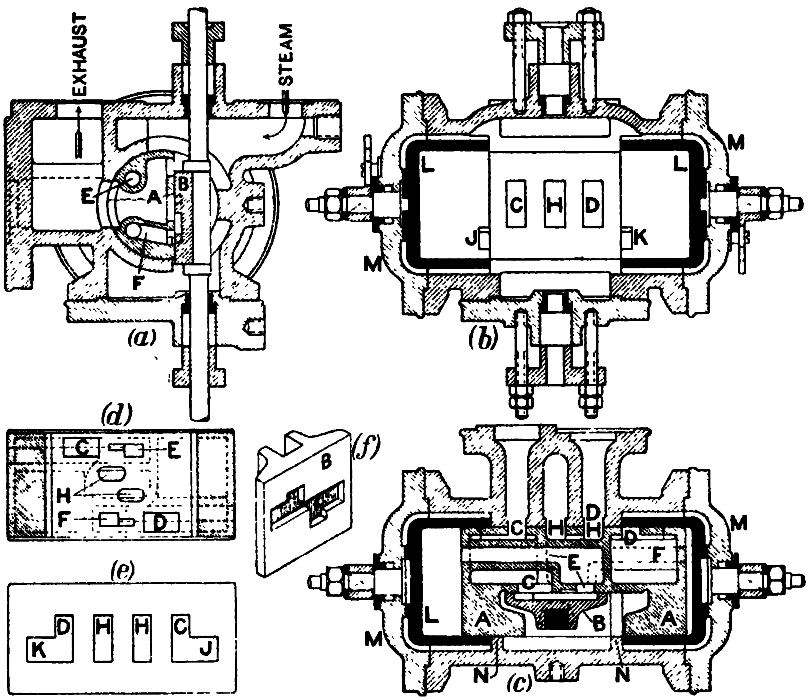


FIG. 188.—Valve chest and steam valves of Weir's feed pump.

valve, and its upper and lower edges cut off steam entering the main ports C and D leading to the bottom and top of the cylinder. The main valve projects beyond the port face of the valve chest and fits into cylindrical caps L in which the valve reciprocates. These caps are held in position by the end covers M and by ribs N cast on the chest. The flat back of the main valve contains the ports C, D, E, and F with the exhaust ports H in the centre.

The operation of the valves during a double stroke of the pump is as follows:—When the piston is at the bottom of its stroke the main valve is in the right hand position as shown at (*c*), the auxiliary valve is at the bottom of its stroke, and the ports C and passages leading to the bottom of the cylinder are open to steam which forces the piston

upwards. When the piston reaches half-stroke the tappet motion (Fig. 187) begins to operate and the auxiliary valve B begins to rise. At about three-quarters stroke the auxiliary valve closes the port C leading to the bottom of the cylinder and the remainder of the upward stroke of the piston is performed by the expansion of the steam already shut into the cylinder below the piston, or by more steam admitted through the by-pass to be described presently. By the time the piston reaches the top of its stroke the auxiliary valve has opened the port F to steam and the port E to exhaust and the main valve has been thrown over to the left, but before completing its stroke the main valve cuts off the exhaust through E and the remaining enclosed steam acts as a cushion and prevents the main valve from striking the end cover. The main valve being now in the left hand position the port C which admitted steam to move the piston up is open to exhaust and the port D leading to the top of the cylinder is open to steam.

The action described above also takes place on the down stroke; the piston moves half its stroke before beginning to move the auxiliary valve, which again cuts off steam at about three-quarters of the stroke. The remainder of the stroke is performed by the expansion of the steam in the cylinder or by fresh steam admitted through the by-pass.

Under certain conditions the pump will not complete its stroke by the expansion of the steam in the cylinder; for instance, if the pump were started with the cylinder cold, the steam would condense rapidly and fall below the pressure necessary to move the piston. In such circumstances steam is admitted through ports J and K, in the caps L, which are brought opposite extensions J and K of the ports C and D in the main valve by turning the caps. This turning of the caps is done by hand from the outside by the arrangement provided at each end of the valve chest. The by-passes are closed after the pump is properly started.

130. Injectors.—The *injector* is a simple appliance for feeding a boiler with water by the direct use of steam from the same boiler or even of steam of a lower pressure as in the exhaust steam injector which utilizes the steam discharged by a non-condensing engine. The injector was invented by Giffard, a French engineer, in 1858.

A well-known modern form of injector is illustrated by Fig. 189, which is a longitudinal section of Holden and Brooke's automatic restarting injector.

Steam enters at A and flows into the steam cone B. In rushing across the opening at the lower end of B a partial vacuum is formed around that opening and water entering at C mixes with the steam in the combining cone DE, the steam being condensed. This condensation of the steam augments the vacuum and increases the flow of water.

The steam rushes through the steam cone with a high velocity and a very considerable velocity is given to the water in the combining cone DE which converges towards E where the passage is narrowest. At this point the velocity of the water, and therefore its kinetic energy, is greatest, but its pressure energy and therefore its pressure is not sufficient to overcome the pressure in the boiler.

Below E is a diverging cone EF which reduces the velocity and

therefore, by the well-known theorem of Bernoulli, increases the pressure of the water, the increase of pressure being sufficient to enable it to enter the boiler. In order that the injector may act properly there must be a more or less definite relation between the quantities of steam and water entering the injector, the relation being different for different steam pressures. The ratio of steam to water is greater the lower the pressure of the steam.

The relation between the quantities of steam and water is regulated by turning the handle H attached to the top of the spindle K. This spindle can only rotate, axial motion being prevented by the collar L. The lower part of the spindle has a screw of rapid pitch formed on it which works in a nut which is in one piece with the steam cone B. This nut and the steam cone are prevented from rotating by the feather key N. It follows that any rotation of the spindle K will raise or lower the steam cone B.

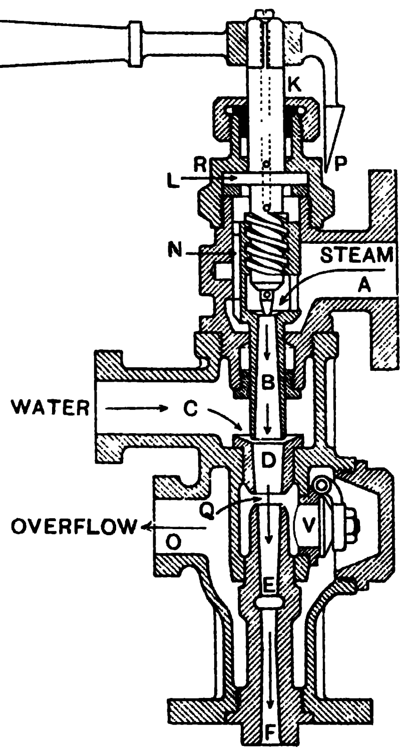


Fig. 189.

The lower end of the spindle forms a steam valve whose seat is the entrance to the steam cone. The lower end of the steam cone forms a water valve whose seat is the entrance to the combining cone. Hence a downward movement of the steam cone increases the supply of steam and diminishes the supply of water. Conversely, an upward movement of the steam cone diminishes the supply of steam and increases the supply of water. For any particular steam pressure there is a particular position for the steam cone which will give the proper mixture of steam and water, and this position is obtained by turning the handle H until the pointer P is opposite to a point on a scale engraved on the ring R which indicates the steam pressure.

The combining cone is interrupted at Q and this gap communicates with the overflow branch O through the flap valve V. There is also a lateral opening in the combining cone at E which is in constant communication with the overflow passage. When the injector is working normally, the steam and water being in proper ratio, there is no escape of steam or water through the overflow passages.

There is a stop valve on the steam pipe which leads from the boiler

to the branch A, and assuming that the handle H has been properly set the injector is started or stopped by operating this valve. In starting, the steam blows through, opening the vaive V and rushing through the overflow passages until the water arrives in sufficient quantity when, a partial vacuum forming in the combining cone, the valve V closes automatically. Also, should the action of the injector be interrupted through a temporary derangement of the steam or water supply, the injector will restart automatically, the valve V opening and the action just described for starting taking place.

A feed check valve is of course fitted at the boiler end of the pipe which leads from the lower end of the injector to the boiler.

As a heat engine for pumping water the injector has a very low efficiency, but when acting as a feed pump and there is no waste at the overflow nearly all the heat taken from the boiler in the steam is returned in the feed water.

A theory of the injector is discussed in Art. 291, p. 406.

131. Feed-Water Heaters.—Apart from the question of economy of fuel due to heating the feed-water before it enters the boiler, this practice is of great benefit to the boiler considered as a structure because it reduces the range of temperature between different parts of the boiler, and in consequence diminishes the stresses due to unequal expansion. Also, the hotter the feed water the smaller is the amount of heat which must be given to it in the boiler, and therefore the more rapidly will it be evaporated, and this more rapid evaporation may be accompanied by a quicker circulation of the water in the boiler which makes the heating surface more effective.

There are two principal classes of feed-water heaters: (1) Those which take the heat required from the waste furnace gases. Heaters of this class are called *economizers*. (2) Those which take the heat required from steam, which is generally, (*a*) the exhaust steam from non-condensing engines, but it may be, (*b*) steam which has done work in one or more stages in a multi-stage expansion engine but has not been through all the stages, or it may be partly (*a*) and partly (*b*). Also, (*c*) the steam used in the heater may be live steam direct from the boiler, in which case the heater is called a *a live steam feed-water heater*.

132. Gain due to Feed-Water Heating.—Let h_1 = heat of 1 lb. of water entering heater; h_2 = heat of 1 lb. of water leaving heater; H = total heat of 1 lb. of steam at boiler pressure. Then gain per cent. due to heating the feed water is $\dfrac{100\,(h_2 - h_1)}{H - h_1}$.

If t_1 and t_2 denote the inlet and outlet temperatures respectively of the water in the heater then it will be sufficiently accurate for practical purposes to substitute t_1 and t_2 for h_1 and h_2 respectively if the temperatures are Centigrade, and $t_1 - 32$, and $t_2 - 32$ if the temperatures are Fahrenheit.

The gain stated above is based on the assumption that the heat used in the heater would otherwise be wasted.

When the feed-water is heated by live steam direct from the boiler it would seem that, apart from the advantages due to using hot feed-water mentioned in the preceding Art., there can be no saving of fuel by this method of heating the feed water. In fact it would appear to

be a case of "robbing Peter to pay Paul." This question has been frequently discussed and careful tests have been made, but the evidence so far is conflicting. In a paper in the *Proceedings of the Institution of Mechanical Engineers* for 1908 Professor J. Goodman and Mr D. R. Maclachlan described careful tests of a boiler without a feed-water heater and with a live steam feed-water heater which appeared to prove that there was absolutely no saving in fuel due to the use of the heater. On the other hand careful tests reported by Professor A. H. Gibson in the *Transactions of the Institution of Engineers and Shipbuilders* in Scotland for 1911 appear to show that there was a saving of 5 per cent. under a light load and 8 per cent. under a heavy load due to the use of a live steam feed-water heater.

133. Green's Economizer.—The feed-water heater knows as *Green's economizer* utilizes the otherwise waste heat of the flue gases and has

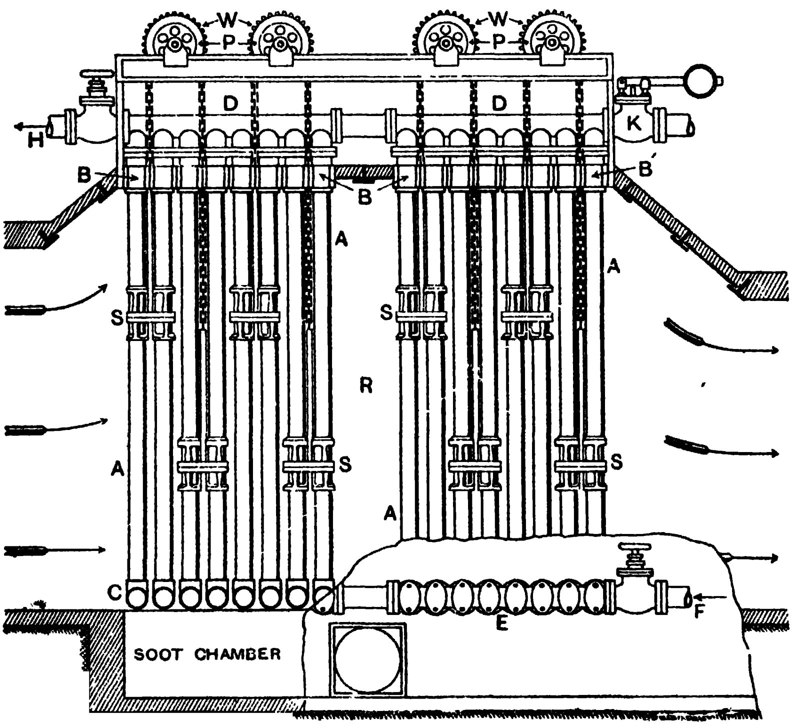


FIG. 190.—Green's economizer.

been very extensively used for stationary boilers, especially those of the Lancashire type. This economizer, which is illustrated by Fig. 190, consists of a large number of vertical pipes or tubes A placed in an enlargement of the flue between the boiler or boilers and the chimney. The tubes are of cast iron 9 feet long, $4\frac{9}{16}$ inches external diameter, and $\frac{7}{16}$ inch thick. The economizer is built up of transverse sections, a section consisting generally of six or eight vertical tubes jointed to

horizontal pipes or boxes B and C at top and bottom. The top boxes B of the different sections are connected to the pipe D, while the bottom boxes C are connected to the pipe E. The pipes D and E are on opposite sides and are outside the brickwork enclosing the economizer. The pipe E has outside openings in it opposite to the bottom boxes, giving access to these boxes for cleaning purposes, the openings being closed by the oval-shaped covers shown. The feed water is pumped into the economizer at F and from the pipe E passes into the bottom boxes C, then up through the tubes A into the top boxes B from which it is led by the pipe D to the pipe H which leads to the boiler or boilers.

At the end of the pipe E opposite to the feed inlet there is a blow-off valve through which any mud or sediment deposited in the bottom boxes may be discharged. Also, at the end of the pipe D opposite to the feed outlet there is a safety valve K loaded to blow off at a pressure a little higher than that of the boiler safety valves.

It has been found to be very essential that the vertical tubes be kept free from deposits of soot which greatly reduce the efficiency of the economizer. For this purpose each tube is provided with a scraper S. The scrapers of two adjoining sections of tubes are grouped together and coupled by rods and chains to the adjacent group of scrapers. The chain passes over a pulley P so that one group of scrapers balances the adjacent group. The pulley P of each chain is connected to a worm wheel W which is driven by a worm on a longitudinal shaft not shown in Fig. 190. The worm shaft is driven through gearing which automatically reverses when the scrapers have reached the top or bottom ends of the tubes. The scrapers are kept in motion continuously while the economizer is in use. The speed of the scrapers is slow, about $2\frac{1}{2}$ feet per minute.

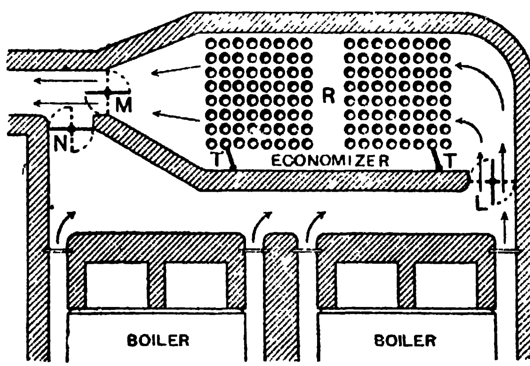


FIG. 191.

The joints of the tubes and the boxes are slightly conical, turned and bored, and are forced together, metal to metal, by powerful hydraulic machinery. The sides of the top boxes are planed so that when bolted together they form an air-tight roof for the economizer.

The temperature of the feed water entering the economizer should not be less than about 35° C. (95° F.), otherwise there is a danger of corrosion due to the moisture in the flue gases being deposited on the cold tubes.

By-pass arrangements for the flue gases and feed-water must always be provided so that the economizer may be put out of action when necessary. Fig. 191 shows an arrangement of the flues of an

economizer for two Lancashire boilers. When the economizer is in use the dampers L and M are open while N is closed. By closing the dampers L and M and opening the damper N the flue gases are sent direct to the chimney and the economizer is put out of action, the feed water being of course at the same time sent direct to the boilers.

When an economizer consists of more than 96 tubes it is usual to arrange the tubes in groups with a passage between adjacent groups for inspection purposes. In Figs. 190 and 191, R is such a passage. It is also advantageous to provide a passage at one side, this passage having two hinged cast iron deflectors T shown in Fig. 191.

134. Surface Condenser Type of Feed-Water Heater.—The most common form of feed-water heater is that in which the water is heated by steam which does not come into direct contact with the water but is separated from it by the walls of metal tubes. The steam used is preferably exhaust steam which would otherwise be wasted. In some designs of this type of heater the steam passes through the tubes and the water to be heated surrounds them, but the heater is more effective if the steam surrounds the tubes and the water is pumped through them. Also, in the former case the casing containing the tubes is under the same pressure as the water which is the full boiler pressure, while in the latter case the casing is only under the pressure of the exhaust steam.

For convenience of cleaning, the tubes should be straight and there should be provision for their free expansion and contraction with changes of temperature, and this should preferably be made without the use of glands and stuffing boxes.

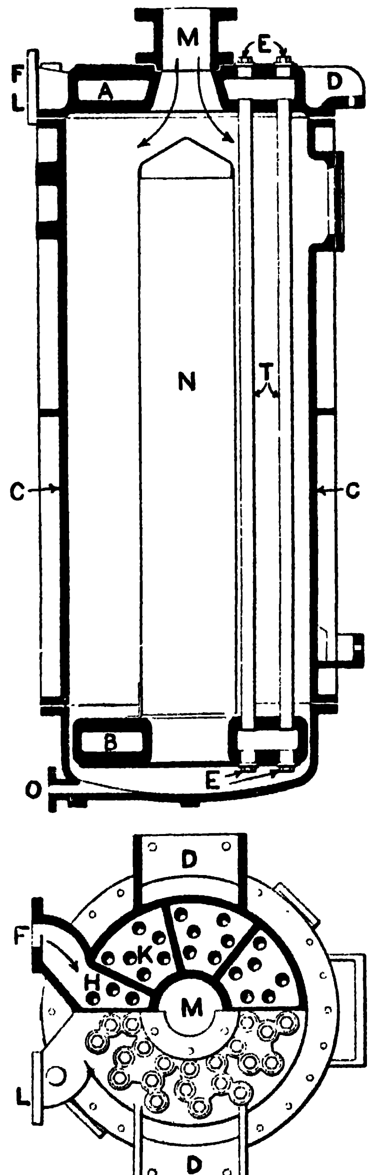


FIG. 192.

A good design of surface feed-water heater is shown in Fig. 192 which illustrates *Weir's multiflow feed-water heater.* In the heater

shown there are forty-two solid drawn copper tubes. For the sake of clearness only two tubes T are shown in the upper view but they are all indicated in the lower views. The tubes are expanded at their ends into cast iron headers A and B, of which A forms the top cover of the cylindrical casing C. The bottom header B is suspended from the top header A by the tubes and is free to move inside the shell when the tubes expand or contract. The top header A has projections D cast on it for supporting the heater.

The headers are divided into compartments by radial ribs, and six tubes enter each compartment, except the compartments at the inlet and outlet in the top header which take three tubes each. Opposite to the ends of the tubes there are holes in the headers fitted with screwed brass plugs E to permit of the fixing or renewal of the tubes.

The water to be heated enters at F into the compartment H in the top header and passes down three tubes to a compartment in the bottom header from which it rises through the three adjacent tubes into the compartment K in the top header. From K the water passes downwards through three tubes to a compartment in the bottom header and so on up and down through three tubes at a time until after going seven times down and seven times up it arrives in the compartment adjacent to H in the top header which it leaves at L on its way to the boiler. A valve casing attached to F and L contains the by-pass valves.

The heating steam enters at M and is deflected and distributed over the tubes by the hollow piece N. The condensed steam is drained away through a valve at O.

The heater is placed on the delivery side of the feed pump and is therefore between the feed pump and the boiler.

135. Weir's Direct Contact Feed-Water Heater.—The feed-water heater shown in Fig. 193 has been very extensively used on board ship. The feed water is heated by direct contact with steam which is taken from the low-pressure receiver of the main engines and also from the exhaust of the auxiliary engines such as feed-pumps, electric light engines, etc. The steam for heating is led by the pipe A to the non-return valve B through which it passes into the interior of the heater. The water to be heated is pumped in at C and passes through the spring-loaded valve D in a thin conical sheet and is at once heated by direct contact with the steam. The water is further broken up into fine spray by passing through the conical spray piece S, thereby bringing the greatest possible surface into contact with the steam. A perforated cylindrical waist-piece U forming an annular steam space round the heater further ensures the uniform mixing of the steam and water.

The heated water and condensed steam, at the boiling temperature corresponding to the pressure, drop into the lower part of the heater in which there is a float E, a water-tight pan open at the top. This float is suspended on two forked levers so that it remains vertical as it rises and falls. The spindle of the top lever is carried through the door at one end and the extended spindle carries outside the heater a lever and weight which balance the float. The float is always full of water, and the weight is adjusted to balance when one half of the float is immersed in water.

The weight lever is connected by a rod to another lever which actuates the throttle valve F and controls the supply of steam to the feed pump which takes the water from the heater at H. In this way the level of the water in the heater is kept practically constant and the pump is kept full of water. The steam from the boiler to work the feed-pump enters the throttle valve at M and leaves at N.

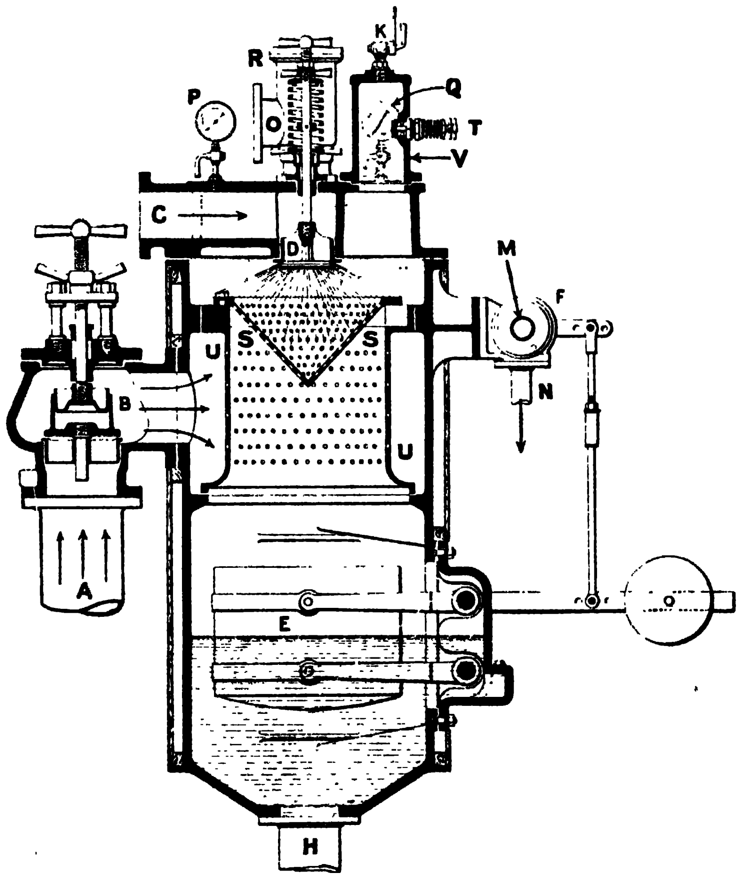


FIG. 193.—Weir's direct contact feed-water heater.

As the pressure in the heater is generally much less than that of the entering water, the effect of this lowering of the pressure, and sudden heating of the water, is to liberate the air in the water, and this is removed to the condenser or to the atmosphere by a small cock K on the air vessel V placed on the top of the heater. The feed water is thus rendered non-corrosive.

A pressure gauge P shows the pressure of the entering water which is fixed by adjusting the spring over the valve D. Another pressure-gauge Q shows the pressure within the heater. R is a relief valve

whose outlet is at O. T is an atmospheric valve which prevents the formation of any considerable vacuum in the heater.

Feed-water heaters of the type described above, not being under pressure, are placed as high as convenient above the feed pump so that the latter may deal more effectively with the hot water from the heater which is on the suction side of the pump.

136. Feed-Water Filters.—The condensed steam from a condensing engine forms nearly the whole of the feed-water for the boilers of such an engine. This condensed steam has in it a considerable quantity of oil which has been used in lubricating the internal parts of the engine. Since oil has an injurious effect on the boilers it is usual to pass feed water containing oil through a filter, two examples of which will now be described.

In the filter shown in Fig. 194 the water enters at L from the feed pump and passing downwards in the direction of the arrows flows through the filtering material H into the chamber D and then upwards through the valve B to the outlet O on its way to the boiler. The filter proper consists of a number of perforated diaphragms which together form deep corrugations around the chamber D. The filter-medium H is a textile fabric made in the form of a sleeve which is drawn over the corrugations and by means of strong twine and with slip knots is pulled in between the corrugations and then tied.

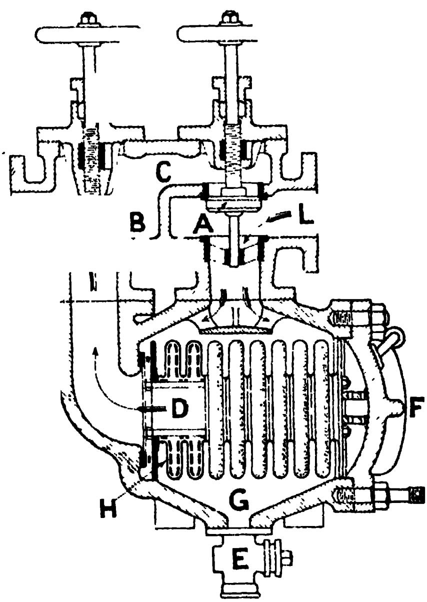


FIG. 194.

The valve A has two faces and controls the inlet to the filter and the by-pass C to the boiler. As shown the by-pass is closed. By shutting down the valve A and closing the valve B the filter is shut off and the feed-water passes through C directly to the boiler.

The filter is cleaned by blowing through as follows. First shut down the inlet valve on its lower seat and then shut the outlet valve B. Open the drain cock E and open the valve A slightly. This will cause a sufficient rush of water over the filtering material to wash it on the outside. Next shut down the valve A and open B. This will cause a rush of water from the boiler through the filtering material and through the drain cock.

To renew the filtering part, the filter being shut off, open the drain cock E to empty the filter chest G. The door F may now be taken off and the filtering part replaced by a spare one.

One form of the *Harris feed-water filter*[1] which has been very largely

[1] Made by Harris Patent Feed-Water Filter (1910), Ltd., Newcastle-on-Tyne.

used is shown in Fig. 195. The body of the filter surmounts the valve chest, forming with it one casting. The body, usually cylindrical, contains a pile of gun-metal grids. The form of these grids is more clearly shown at (*a*). Each grid consists of an outer rim A and an inner rim B. The upper and lower edges of these rims are machined and within each there is a shallow recess. The rims are connected by

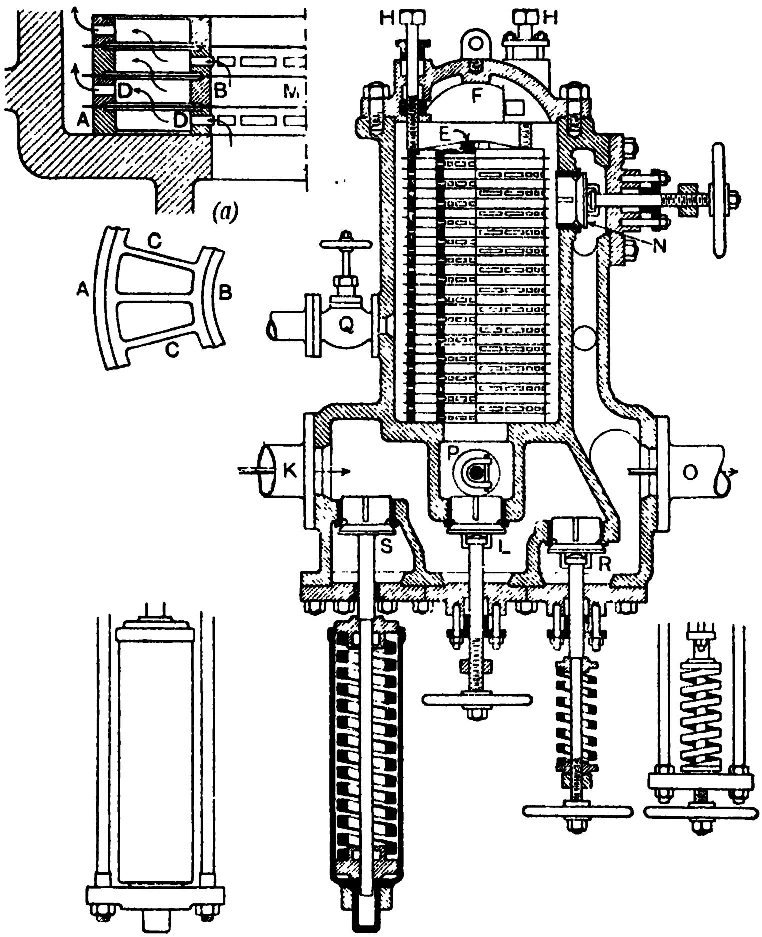


FIG. 195.—Harris feed-water filter.

radial arms C. Ports D are provided in the inner and outer rims respectively of alternate grids.

The bottom grid rests on a ring of jointing material on the machined bottom of the filter body. The ports of this grid are in its inner rim. In the upper recess of this grid is laid a disc of tinned copper-wire gauze, and above that a circular filter cloth, of closely-woven flax fibre, with a central hole, which overlaps the machined edges of the inner

and outer rims. The next grid, the ports of which are in the outer rim, with its gauze discs is placed upon the cloth which is thus held between both rims of the two grids and therefore makes a watertight joint. The third grid, with ports similar to the first one, together with its gauze discs and filter cloth then follows and thus the filtering portion is built up and is surmounted by a strong cover E. The body cover F is then placed and secured in position and the pile of grids compressed by the holding-down screws H.

The action of the filter is as follows. The dirty water entering at K from the feed pump is admitted by the valve L to the chamber M formed by the inner rims of the pile of grids. It then passes through the ports of the inner rims, through the filtering medium, and out by the ports in the outer rims, as is clearly shown by the arrows in the detail illustration (*a*). The filtered water then passes through the valve N to the outlet O and thence to the boiler.

As deposit accumulates upon the cloths the pressure required to force the water through the filter increases. Two pressure gauges are provided so that the rise of pressure can be observed, and should it exceed a certain predetermined amount the spring-loaded by-pass valve R opens and allows the water to pass direct to the boiler.

When dirty the filter is cleaned as follows. The inlet valve L and outlet valve N are closed and the boiler supplied for the time through the by-pass. The steam valve Q is opened and steam passed into the filter body to heat the contained water and thus soften and loosen the deposit. The sludge valve P is then opened, and then by opening the valve N water under boiler pressure passes the reverse way through the cloths and carries off the deposit through the sludge valve P. The sludge and steam valves are then closed, the inlet valve opened and the cleaned filter is once more in operation.

The spring loaded relief valve S is only fitted when the filter has to pass Board of Trade survey requirements.

137. Feed-Water Regulators.—In boilers of small water capacity it is very necessary that the water level be maintained as nearly as possible constant, which means that the rate at which the feed water is supplied must be the same as that at which it is evaporated in the boiler.

A number of appliances which automatically regulate the feed-water supply are in successful operation. In most of these *feed-water regulators* use is made of a float, the rise and fall of which operates, directly or indirectly, a valve in the feed pipe. In several designs the regulator is inside the boiler and the float acts on a feed-water valve in much the same way as the float in an ordinary storage cistern.

In the White-Forster feed-water regulator, now to be described, the apparatus is placed outside the boiler and the float operates a valve in a piston, which, under steam pressure actuates the valve in the feed pipe. This feed-water regulator is shown in Fig. 196. (*a*) is a vertical section, (*b*) an elevation, and (*c*) a plan. At (*d*) the greater part of the piston is shown to a larger scale.

The chamber C containing the float F is connected to the steam space in the boiler by a pipe with a valve at S and to the water space by a pipe at W. The piston P, which is worked by steam pressure in

the small cylinder Y, is connected to the feed regulating cock K through the piston rod R, the lever L, the link N, and the lever H as shown. The steam to the piston P is governed by the small valve V which is worked by the float F. The steam to this valve passes through

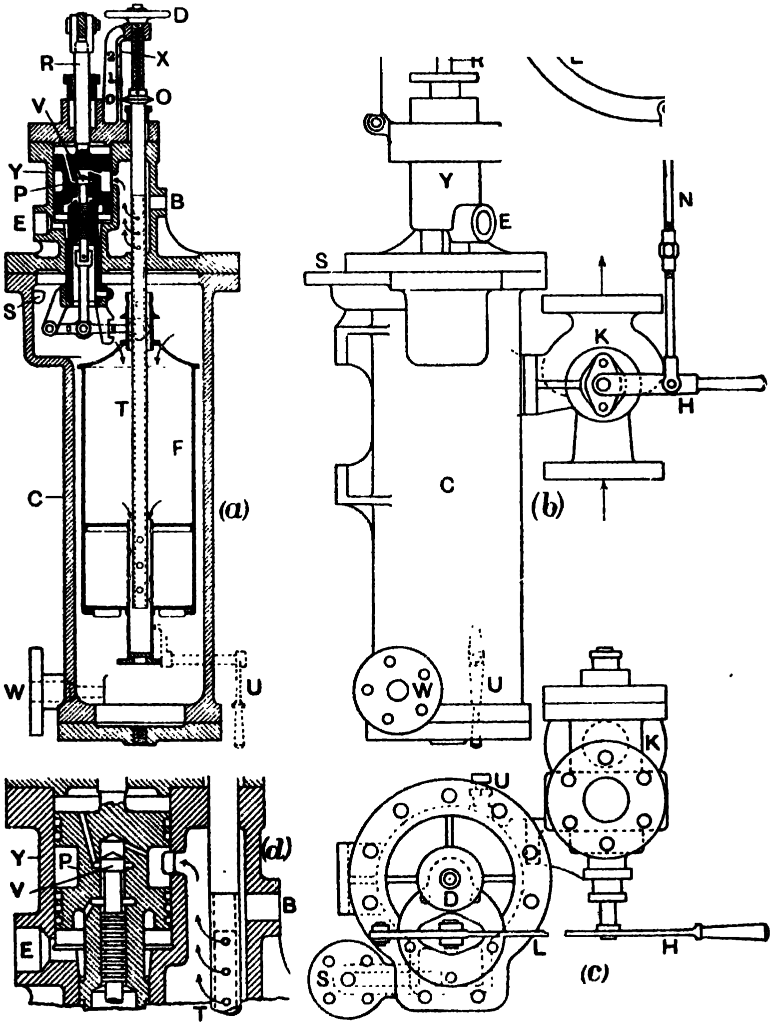


FIG. 196.– White-Forster feed-water regulator.

the float, entering at the top and leaving at the bottom by the tube T; this prevents water accumulating in the float, relieves it of all pressure and therefore enables it to be made of thin sheet brass and without balance weights. The lower part of the float contains water up to the level of the lower end of the tube T; hence the buoyancy of the float

and the level of the water in the float chamber depend on the position of the tube T.

If the water level in the boiler falls so does that in the float chamber and the float lowers the valve V, which admits steam to the top of the piston P, through the passages shown, and drives it down until the steam port is again closed. On the other hand, if the water level in the boiler rises, the float raises the valve V and connects the top of the piston with the exhaust pipe at E and the unbalanced pressure on the bottom part or small diameter of the piston forces it up until the port is again closed. The piston P therefore copies exactly the motion of the valve V and float F.

To raise or lower the water level in the boiler the steam valve at S is closed and a blow-through valve at B is opened; this drowns the float and the tube T is then raised or lowered the desired amount by means of the hand wheel D. S is then opened and as soon as dry steam appears at B the valve there is closed and the regulator is again in working order. The effect of this is to alter the amount of water in the bottom of the float and so alter the buoyancy of the float and the level of the water in the float chamber.

The water level for which the regulator is set is indicated by the pointer O and index or scale X.

By means of the handle on the lever H attached to the feed-water cock K the free movement of the outside gear may be tested. At the bottom of the float chamber there is another lever U so arranged that the free action of the float and its connections may be tested when necessary.

Another form of feed-water regulator which has been highly successful is shown in Fig. 197. This is the *Crosby feed-water regulator*, which is a very ingenious apparatus consisting of two principal parts, the regulating valve 1 through which the feed-water pases on its way from the feed pump to the boiler, and the power producer or *operator* 2 which actuates the regulating valve as required by the varying evaporation in the boiler.

The operator is really a small boiler hermetically sealed and containing its own water supply, forming a closed system which is entirely independent of the main boiler.

In the lower part of the operator there is a partition 3, the space beneath which is in communication with the steam space of the boiler through the pipe 4 and also with the water space through the pipe 5, the water being cooled in vessel 6.

The space above the partition 3 contains water and steam and the pressure of this steam, which is never high, depends on whether steam or water is in contact with the under face of 3.

A rise or fall in the level of the water in the boiler causes a reduction or an increase in the pressure of the steam in the operator.

When the steam pressure in the operator rises above the normal the pressure acting on the column of water in the coil 12 and pipe 7, which lead to the upper face of the diaphragm 8 (Section *c*) contained in the diaphragm chamber 9, overcomes the resistance of the adjustable spring 10, and increases the opening of the regulating valve 1 and allows an increased amount of feed-water to pass to the boiler. When

the pressure in the operator falls below the normal the reverse action takes place.

The sensitiveness of the operator to changes of water level in the boiler is increased by means of the corrugations 11 on the partition 3 (Sections *a* and *b*).

The temperature of the water upon which the steam acts in the

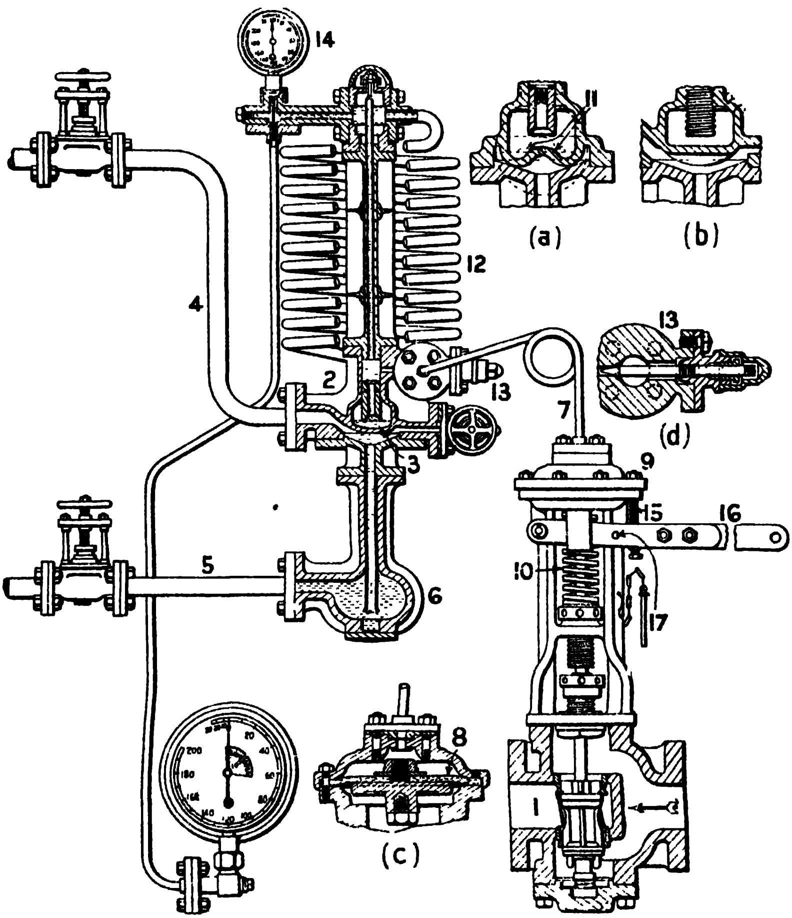


FIG. 197. - Crosby feed-water regulator.

operator is kept down, and the steam pressure is prevented from rising unduly by the large cooling surface presented by the coil 12 so that there is an entire absence of "hunting" and the feeding of the boiler is regular and continuous and in proportion to the rate of evaporation.

The return of the water to the heating surface is under control by means of the needle valve 13 (Section *d*) which is adjustable externally thus allowing a correction to be made on it should the position in

which the operator has to work be either cooler or hotter than the normal.

The regulating valve 1 being a double-seated valve gives a large area of opening with a comparatively small movement.

The diaphragm 8 is made of specially thick rubber reinforced by canvas and is protected from the effect of heat by the cool water immediately above it.

A pressure gauge 14 is fitted to the operator as an indicator showing at all times the position of the regulating valve. A supplementary gauge may be added at a lower level, as shown, when the upper gauge cannot be readily seen by the stoker. A jacking screw 15 is provided on the regulating valve lever 16 by means of which a minimum opening of the valve and therefore a minimum feed is fixed if required.

The apparatus works very rapidly and is sensitive to changes in the rate of evaporation which are scarcely apparent in the gauge glass.

The whole apparatus can be put out of action instantly by pulling down the lever 16 and locking it with the pin 17. The valve is then in its full open position.

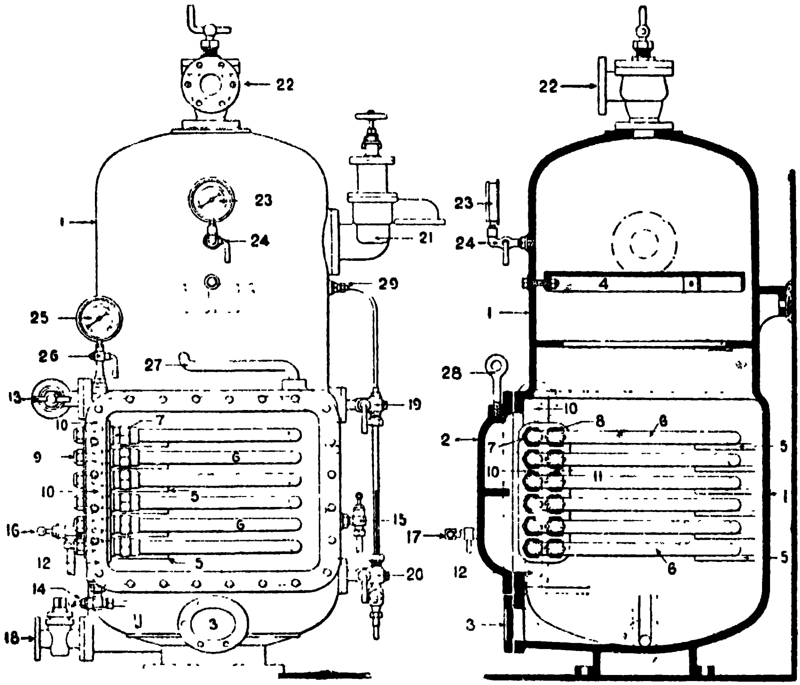
138. Evaporators.—In an economical steam power plant the steam after doing its work in the engine is condensed and returned to the boilers. The same water is therefore used over and over again and if there were no loss of steam or water anywhere no new feed-water would be required for the boilers. But there is always a loss going on due to leakage at various points and to blowing off at the safety valves. This loss may generally be taken at from 4 to 8 per cent. of the total feed-water.

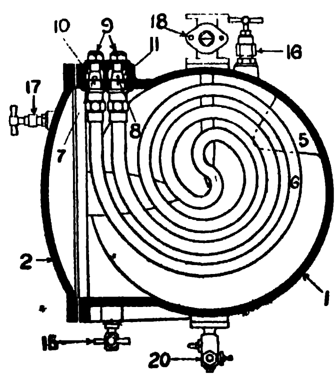
When there is no fresh water suitable for making up the boiler feed the water available must be treated in some way in order to purify it. On board ship, where sea-water only is available, the make-up feed-water is obtained by evaporating sea-water in a vessel called an *evaporator*, the steam produced from the salt water being passed either into a feed-water heater or into the condenser where on being condensed it joins the main feed-water.

The heat required to evaporate the water fed into the evaporator is supplied by steam taken direct from the boilers. This heating steam passes through coils of tubes which are surrounded by the water in the evaporator. The water resulting from the condensation of the heating steam drains into the hot well and is returned to the boilers. To prevent a too great accumulation of salt left from the water evaporated a brine pump is provided which removes, at a constant rate, a certain amount of the hot concentrated sea-water from the evaporator and returns it to the sea.

A highly approved form of evaporator is shown in Fig. 198, which illustrates Weir's evaporator. An index of parts is also appended. It may be added that the shell is of cast iron and that the heating surface is formed of heavy solid drawn copper tubes arranged in elements, each element consisting of a single tube coil with special hollow steel couplings on its inlet and outlet ends. These hollow couplings pass through a steam chamber cast on the side of the evaporator and are secured in place from the outside by means of a cap nut on the end of each coupling.

The outlet opening (8) from the tubes is of smaller diameter than the inlet (7), with the exception of that in the lowest tube, which is a drain tube, and through which any steam passing through the smaller outlets from the tubes above is again returned through the evaporator along with the water of condensation and drained to the hot well. By





1. Shell of evaporator.
2. Main door of evaporator for withdrawing tube coils (6).
3. Hand cleaning door.
4. Baffle plate or deflector.
5. Supports for coils (6).
6. Evaporating tube coils.
7. Inlet steam couplings for coils (6).
8. Drain outlet couplings for coils (6).
9. Coupling nuts for 7 and 8.
10. Inlet steam header.
11. Drain header.
12. Drain collecting pocket.
13. Inlet valve for steam coils (6).
14. Valve for drain from coils (6) to hot-well.
15. Feed check valve.
16. Brine valve.
17. Salinometer valve.
18. Cock for blowing off to sea.
19. Top cock for water gauge.
20. Bottom cock for water gauge.
21. Safety valve.
22. Outlet valve for generated steam.
23. Compound gauge for generated steam in shell.
24. Cock for compound gauge (23).
25. Pressure gauge for inlet steam to coils (6).
26. Cock for pressure gauge (25).
27. Swing crane bar for door (2).
28. Eye bolt for supporting door (2) on crane bar (27).
29. Connection from top of water gauge to steam space in evaporator shell.

FIG. 198.—Weir's evaporator.

this means the pressure in the standpipe chamber at the outlet end is always lower than in the chamber at the inlet end, and a constant current is kept up through all the tubes, thus preventing any accumulation of air and water. Each element or tube can be taken out separately for cleaning.

139. Reducing Valves.—The steam supplied to an engine should be as nearly as possible of uniform pressure. With boilers of the Lancashire type or the Scotch marine type, which have a large water capacity, it is comparatively easy to maintain a nearly constant pressure, but with water-tube boilers, which hold a comparatively small quanity of water, it is more difficult to keep the pressure uniform. It is a common practice to work such water-tube boilers at a pressure higher than that for which the engine is designed. The steam is then passed through a *reducing valve* on its way to the engine, the function of this valve being to maintain a constant reduced pressure on the engine side of the valve while the higher pressure on the boiler side may be variable.

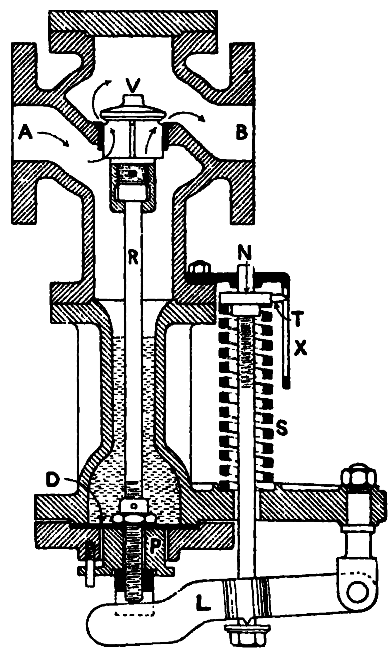


FIG. 199.

There are also many cases in practice where steam of a lower pressure than that in the boiler is required, and steam of the required lower pressure is then obtained by the use of a reducing valve.

The principle upon which all reducing valves work is the principle of the throttle valve. An ordinary stop valve may be made to act as a reducing valve by opening it partially, the reduction of pressure being greater the smaller the opening of the valve, but if the opening is kept constant an increase or decrease of pressure on the supply side will be accompanied by an increase or decrease of pressure on the delivery side.

In a proper reducing valve the amount of the opening of the valve is regulated automatically so as to give a practically constant pressure on the delivery side, although there may be a variation of pressure on the supply side, but the pressure on the delivery side cannot of course exceed the pressure on the supply side.

One form of reducing valve is shown in Fig. 199 which illustrates the Auld type of valve. Steam of the higher pressure enters at A and passing through the valve V leaves at B at the required reduced pressure. The valve V is connected to the loose fitting piston P by the rod R.

Leakage past the piston is prevented by the flexible india-rubber diaphragm D. The diameter of the piston being the same as that of the valve the upward pressure of the steam on the valve is balanced by its downward pressure on the piston. When steam is passing through the valve the pressure of the reduced pressure steam on the top of the valve V is balanced by the force exerted by the spring S acting through the lever L as shown. The force exerted by the spring is fixed by means of the nut N, to which is attached the pointer T, whose position on the index X shows the pressure on the delivery side of the valve.

A fall or rise of pressure at A is followed by a momentary fall or rise of pressure on the top of the valve V which, being forced upwards by the constant force from the spring S, will rise or fall until the pressure above the valve is the reduced pressure for which the spring is adjusted.

Water formed by the condensation of steam in the chamber above the piston protects the rubber diaphragm from the action of the steam and keeps it cool.

The reduction of pressure due to the action of a reducing valve is accompanied by a slight drying or superheating of the steam.

140. Steam Driers or Separators.—When steam supplied to an engine is not superheated it is almost certain to contain water suspended in it. This water may have been carried away, as water, from the boiler, but even when the steam is dry on leaving the boiler there is, with stationary and marine engines, a certain amount of condensation in the steam pipe between the boilers and the engines. Since water carried by the steam is very objectionable in the engine it is a common practice with stationary and marine engines to provide a *steam drier* or *steam separator* near to the engine for the purpose of removing as completely as possible the water from the steam.

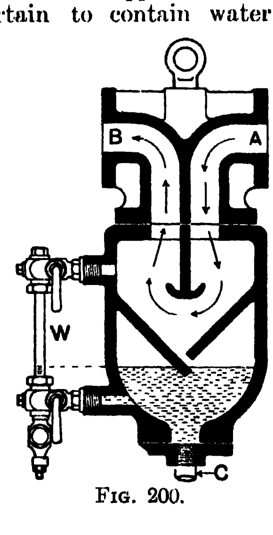


Fig. 200.

There are numerous designs of steam separators but they all consist of a chamber through which the steam passes and in which the current of steam is made to change its direction and throw out the water by reason of its greater inertia.

A simple form of steam separator is shown in Fig. 200 which illustrates a design of Messrs. Alley and Maclellan of Glasgow. The steam enters at A and passing in the directions of the arrows leaves at B. The water is thrown down into the lower part of the separator and is carried away by a pipe C, preferably connected to a steam trap. If a steam trap is not used the water collected by the separator must be discharged periodically by opening by hand a valve in the pipe C. A glass water gauge W exhibits the level of the water in the separator.

In a design of separator suggested by Sir Charles Parsons, and

used on the Cunard liner *Aquitania*, the steam is made to pass through a number of grids of different metals. The aim in this design is not only to separate the water from the steam but to remove any corrosive elements which might attack the metals in the steam turbines by presenting to the steam the metals cast iron, steel, and brass in the separator where the corrosive elements in the steam might spend themselves on the renewable metal grids instead of doing so in the turbines.

141. Steam Traps.—Steam traps are more often accessories to the engine than to the boiler, but it will be convenient to describe examples of them here. The function of a steam trap is automatically to drain away or return to the boiler water resulting from the partial condensation of steam in steam pipes, steam jackets, etc., without allowing any steam to escape.

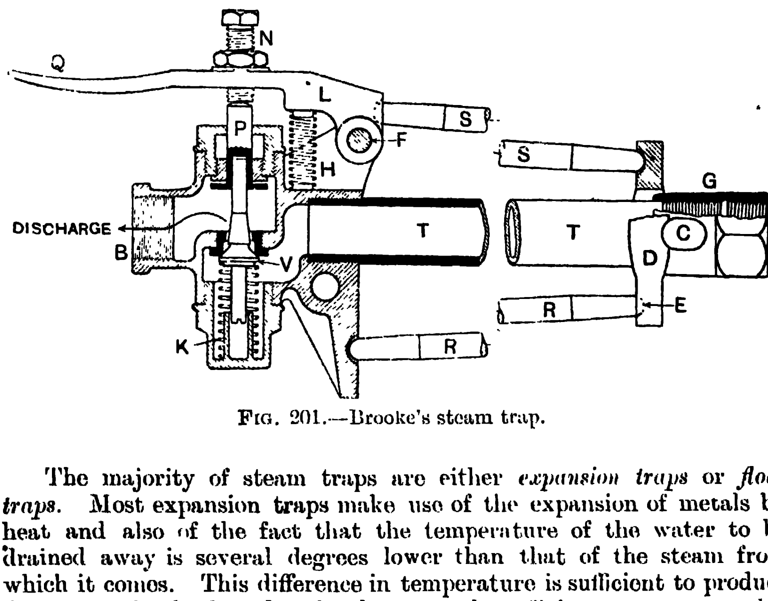


FIG. 201.—Brooke's steam trap.

The majority of steam traps are either *expansion traps* or *float traps*. Most expansion traps make use of the expansion of metals by heat and also of the fact that the temperature of the water to be drained away is several degrees lower than that of the steam from which it comes. This difference in temperature is sufficient to produce differences in the lengths of tubes or rods sufficient to open a valve which allows the water to escape and to close it when the water is discharged. In float traps a closed hollow float in a closed vessel into which the condensed steam is led rises as the water increases, or a bucket in the closed vessel floats until enough water has run into it to sink it. The rising of the float or the sinking of the bucket is made to open the discharge valve and when the greater portion of the water has escaped the falling of the float or the rising of the bucket causes the valve to close.

An example of a simple and efficient expansion type of steam trap is shown in Fig. 201. This is Brooke's steam trap made by Messrs. Holden and Brooke, Manchester. The trap is coupled at A to the drain pipe from the steam pipe or vessel to be drained. The water is discharged through the valve V and is led away by a continuation of

the drain pipe coupled to the trap at B. The expansion element is the brass tube T. When the tube T contains steam the valve V is closed. If water enters the trap the tube T contracts and the projecting pieces C, on opposite sides of the coupling G, pressing on the middle of the rocking piece D cause the latter to turn about the rounded end E of the rod R and push the rod S which moves a distance twice that of C. The motion of the rod S is transmitted to the lever L, whose fulcrum is the pin F, and the tilting of this lever opens the valve V. As soon as the water is discharged steam arrives and the temperature of the tube is raised with the result that the tube expands, the rod S is slackened, and the spring H forces the lever L upwards and allows the valve to close under the pressure of the steam and the

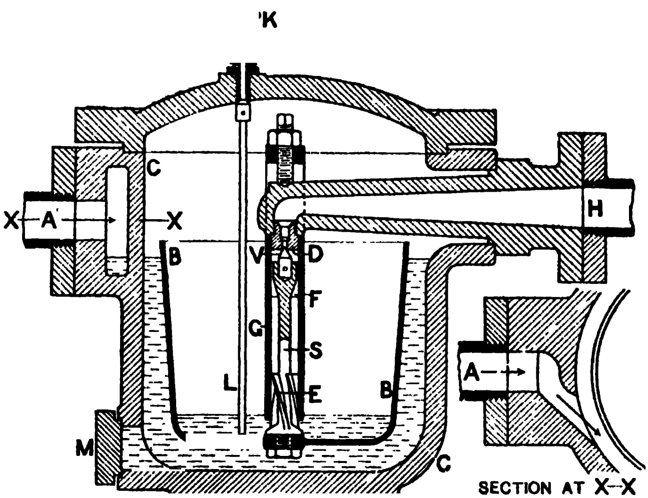


FIG. 202.—"Sentinel" steam trap.

force exerted by the light spring K. The screw N is adjusted so that when the valve is closed the sliding piece P between N and the valve spindle has a slight freedom vertically. To open the valve for the purpose of blowing through at any time it is only necessary to depress, by hand, the extension Q of the lever L.

As an example of an efficient float or bucket type of steam trap the "sentinel" trap, made by Messrs. Alley and Maclellan of Glasgow, may be taken. This steam trap is shown in Fig. 202. The water to be drained away enters the casing C at A. The casing always contains sufficient water to float the bucket B to which is attached the spindle S. This spindle has attached to its upper end the valve V of which D is the seating. The valve and its seating are made of pure nickel. The spindle S has spirally formed vanes on it at E and F which can move easily inside the guide tube G.

When the water has risen high enough in the casing C it overflows into the bucket B which ultimately sinks and the valve V opens. The water is then forced up the guide tube G by the pressure of the

steam and passing through the valve leaves the trap by the outlet H. When the bucket is nearly emptied it again floats and the valve is closed.

The water flowing up through the guide tube impinges on the spirally formed vanes and causes the spindle S and bucket B to rotate. This rotary motion attains its maximum speed as the valve is about to close and by the fly-wheel action of the bucket the valve is reground each time the trap acts. Any roughness on the valve faces caused by grit is burnished over and remedied immediately it occurs, thus ensuring that the trap will keep tight indefinitely. It will be seen from the section at XX that when the trap is blowing off the inrush of water and steam strikes the bucket tangentially, thus accelerating the rotation of the bucket and assisting the regrinding action.

By pressing on the knob K the rod L is depressed and the bucket pushed downwards and the valve opened. The trap may therefore be blown through and tested at any time. The door M gives access for the cleaning out of any sediment which may accumulate in the casing.

As a trap of this design depends for its action on the retention of some water of condensation in the casing, it is necessary where the steam is highly superheated to protect this water from re-evaporation by the insertion of about 10 feet of pipe between the steam range and the trap.

142. Superheaters.—The following references to articles in this work on superheated steam, its properties, applications, advantages, etc., may be given here—Arts. 51 to 54, 143, 144, 173, 280 to 283.

Superheating is effected by passing the boiler steam through an assemblage of steel tubes of comparatively small diameter forming the main part of the *superheater*. There being a continuous passage between the boiler steam space and the interior of the superheater there can be no change in the pressure due to superheating, but its volume increases as its temperature is raised.

The superheater generally receives its heat from the hot gases leaving the boiler furnace but the superheater may be a separate unit having its own furnace independently fired. Generally the independently fired superheater is only used in connection with large boiler installations.

When the superheater is heated by the gases from the boiler furnace it is usual to place it, in the case of Cornish and Lancashire boilers, in the downtake at the back end of the boiler. In locomotive boilers and in marine boilers of the Scotch type the superheater is placed either in large tubes or pipes which take the place of a number of the ordinary tubes, or in the smoke-box or in the uptake.

The forms and positions of the superheaters used with Babcock and Wilcox and with Stirling water-tube boilers are shown in Fig. 98, p. 133, and Fig. 102, p. 137.

In subsequent Arts. Sugden's superheater, and Schmidt's locomotive superheater are illustrated and described.

In nearly all superheaters attached to individual boilers provision must be made to prevent the overheating of the tubes while steam is being raised and when delivery of steam from the boiler is suspended.

The methods adopted for this purpose are—(1) Flooding the super-superheater with water from the boiler, this water being drained out before the delivery of steam from the boiler is commenced or resumed. (2) Diverting the hot gases or stopping their flow over the superheater tubes. (3) Circulating a comparatively small quantity of steam through the superheater, this steam being discharged into the atmosphere.

143. Sugden's Superheater.—A form of superheater which is extensively used in connection with stationary boilers, especially those

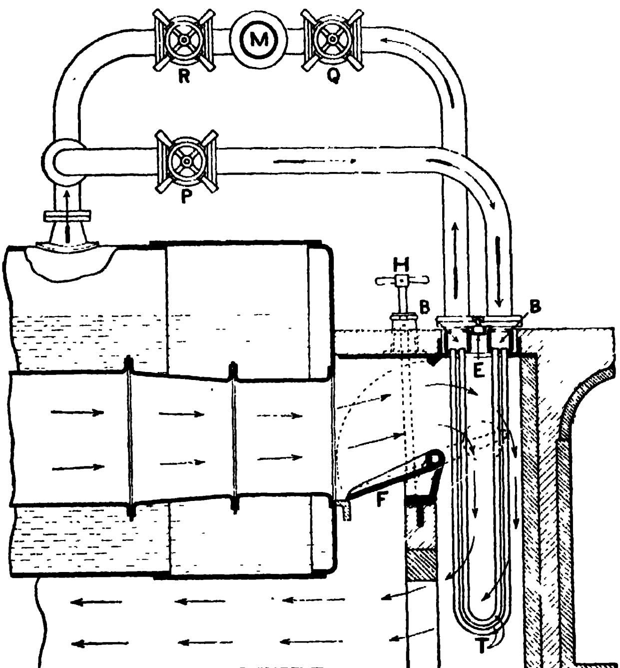


FIG. 203.—Sugden's superheater.

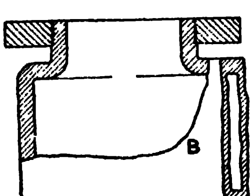


FIG. 204.

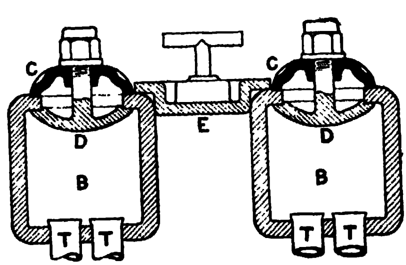


FIG. 205.

of the Lancashire type, is shown in Figs. 203, 204, and 205. This is Sugden's superheater and is shown in Fig. 203 as applied to a

Lancashire boiler. The superheater is placed in the downtake at the back of the boiler where the temperature of the flue gases is generally not less than 550° C. (1022° F.). A superheater of this design is usually constructed to give a superheat of 40° C. to 95° C. (104° F. to 203° F.).

This superheater consists of two mild steel boxes or headers B from which hang groups of solid drawn steel tubes T bent to U-form. The ends of these tubes are expanded into the headers as shown in Fig. 205. The tubes are arranged in groups of four and one pair of headers generally carries ten of these groups or forty tubes in all. Opposite to each group of tubes the headers are provided with oval hand holes having external and internal covers C and D giving access to the ends of the tubes. The space between the headers gives access to the outside of the tubes for the purpose of inspection and cleaning. This space is closed by the covers E. Any tube may be taken out and replaced without disturbing the superheater as a whole.

The branches on the headers for the inlet and outlet of the steam are forged on the headers and have steel flanges screwed on to them as shown in Fig. 204. Steam enters at one end of the rear header and leaves at the opposite end of the front header.

To prevent the overheating of the superheater tubes when steam is first being raised, and in order that the superheater may be put out of action at any time if necessary, a balanced damper F is provided and is operated by the handle H. As shown in Fig. 203, the superheater is in action but by turning the damper F upwards into the vertical position the gases pass directly into the bottom flue without passing over the superheater tubes. By placing the damper in intermediate positions part of the gases will go over the superheater tubes and the remainder will pass directly to the bottom flue and varying degrees of superheat may thus be obtained.

Fig. 203 also shows how the steam pipes may be arranged so as to pass the steam through the superheater or direct to the main as may be required. M is the main steam pipe, P, Q, and R are stop valves. When the superheater is in action the valves P and Q are open and R is closed. When steam is taken from the boiler direct to the main steam pipe M the valves P and Q are closed and R is open.

144. Schmidt's Superheater.—As a good example of a superheater giving a high degree of superheat the Schmidt superheater may be taken. The form of this superheater as used largely on locomotives is shown in Fig. 206. In the upper part of the tube space of the barrel of the boiler there are two or more, generally three, rows of large tubes T expanded into the fire-box and smoke-box tube plates. These tubes have an external diameter of $4\frac{3}{4}$ inches to $5\frac{1}{2}$ inches, but towards the fire-box this diameter is reduced. In each of these tubes is inserted a superheater element consisting of two sets of tubes bent to U-shape forming a continuous double-looped tube *t*. The ends of this element are connected to a header or collecting casting in the smoke-box.

The saturated or wet steam from the boiler enters the header at A and passes into the chamber B which is in communication with the saturated steam end of each superheater element. The steam then traverses the superheater elements to and fro and is delivered into the

chamber C of the header from which it passes to the steam pipes at S, to right and left, leading to the cylinders.

The superheater elements are made of seamless steel tubes, the ends near the fire-box being either screwed into cast steel U bends or they are welded to form the bends as shown in Fig. 206. The open ends of the elements in the smoke-box are expanded into flange blocks F which are secured to the header, each by one central bolt.

As shown in Fig. 206 a separate compartment is formed in the smoke-box for the larger tubes containing the superheater elements, this compartment being provided with a damper D which regulates the draught through these tubes, and therefore controls the amount of superheat. When the steam is shut off from the cylinders by the closing of the regulator valve the superheater is isolated by

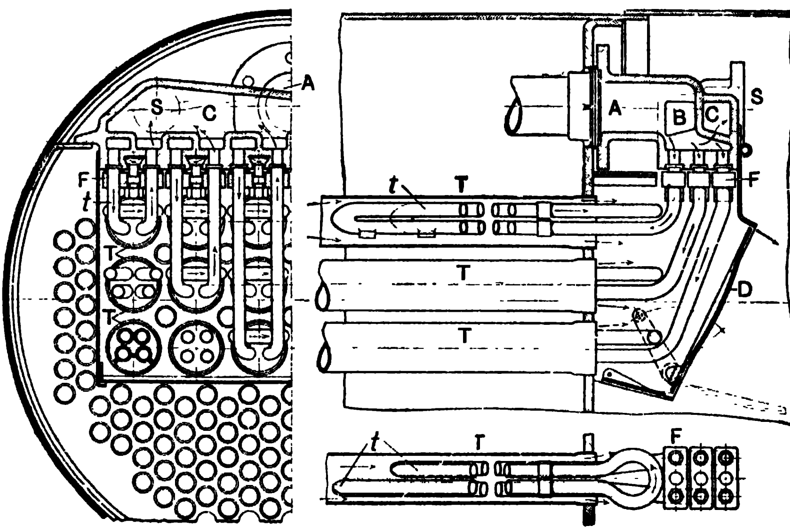


FIG. 206.—Schmidt's locomotive superheater.

closing the damper D, as shown in Fig. 206, and overheating of the superheater elements is prevented.

In the later designs of the Schmidt locomotive superheater the compartment for the superheater in the smoke-box, with its damper or dampers, is dispensed with by the introduction of a system of circulating automatically a portion of steam through the superheater elements when the regulator is closed. Before describing this system reference may be made to Fig. 207 which shows a more recent design of superheater than that illustrated by Fig. 206. In the later design the flange blocks F are loose on the tubes and are bolted against collars H fixed to the tubes. Also the nuts for pulling up the flange blocks are now on the top and out of the way of the current of hot gases and are more accessible than in the older type.

Coming now to the system of circulating the steam through the superheater when steam to the cylinders is cut off. This is carried out

by means of automatic by-pass valves shown in Fig. 208. There are two of these valves placed back to back, one for each cylinder. Pipes J connect the ends of each cylinder with its by-pass valve. Pipes K connect the cylinder steam pipe in the smoke-box to the by-pass valve cover, thus leading steam to the larger face of the piston P which forms the upper part of the valve V. A pipe L is taken direct from the boiler steam space (Fig. 207) to a cock, not shown, on the outside of the smoke-box, and thence to the underside

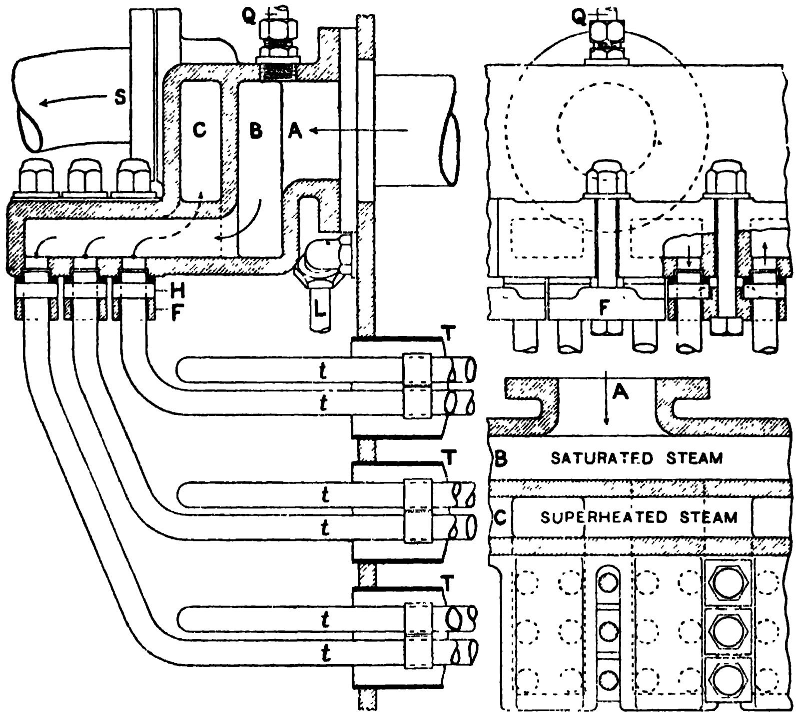


FIG. 207.—Schmidt's locomotive superheater.

of each piston P. From the underside of the piston P of one of the valves a pipe Q is taken to the saturated steam side of the header (Fig. 208). Waste pipes W lead to the atmosphere.

The steam cock mentioned above is opened when steam is being raised and is left open while the engine is in service. When the engine is running with the regulator open, steam is admitted from the cylinder steam pipe through the pipe K and acting on the larger face of the piston P closes the by-pass valve V and the relief valve R. On closing the regulator the boiler pressure steam, always carried by the pipe L when the engine is in service, opens the by-pass valve V and relief valve R thus establishing a connexion between the ends of engine cylinder and the atmosphere through waste pipe W, so that steam from the boiler passing through the pipe L is led by the pipe Q

to the saturated steam side of the header, passes through the superheater and into the steam pipe leading to the cylinder and then by the pipes J through the relief valve R and waste pipe W to the atmosphere.

In the event of the engine standing in mid gear with all cylinder ports closed, provision is made to permit of circulation of steam through the superheater by the introduction of a port U in the valve casing controlled by the piston P. Thus when the engine is standing with regulator closed the piston valve is in the open position and port U is uncovered. The circulating steam is therefore able to escape by port U and pipe W to the atmosphere.

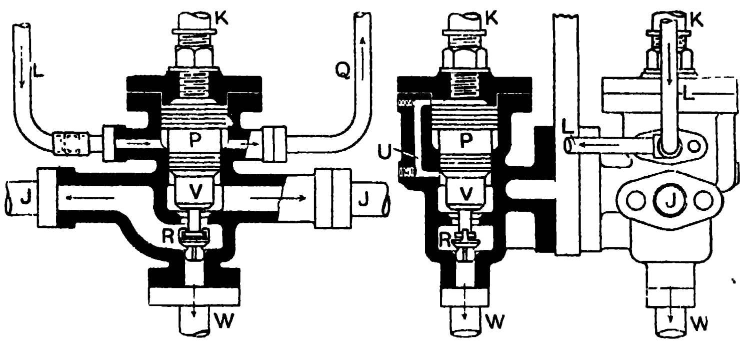


FIG. 208.—Auto by-pass valves for Schmidt superheater.

In the modification of the Schmidt superheater designed by Mr. John G. Robinson, chief mechanical engineer of the Great Central Railway, small jets of steam are automatically blown into the large tubes containing the superheater elements, from the smoke-box end, when no steam is being used by the engine. This spoils the draught through these tubes and prevents the overheating of the superheater elements.

It is common in locomotive practice when superheating is adopted to raise the temperature of the steam to about 340° C. (644° F.). With steam at 180 lb. per square inch pressure by gauge this means a superheat of about 147° C. (265° F.).

145. Thermal Storage.—There are many cases in practice, particularly in electric lighting stations, where the load on the engines is variable, reaching a maximum for two or three hours only every twenty-four hours, the maximum load being very much higher than the mean load. To provide boiler power in the ordinary way sufficient for the maximum load would mean that at light loads the boilers would have to work very inefficiently.

The thermal storage system introduced by Mr. Druitt Halpin was designed to meet such cases. Briefly, this system enables the boilers to work at a more nearly uniform rate by using the surplus steam at light loads to raise the temperature of extra feed-water to the temperature of the steam, this extra feed-water being stored in a vessel generally placed on top of the boiler.

Fig. 209 shows the thermal storage system applied to a Babcock and Wilcox boiler. The feed-water is pumped into the storage vessel at a rate greater than is required by the boiler at the lighter loads and from the storage vessel it flows by gravity into the boiler through

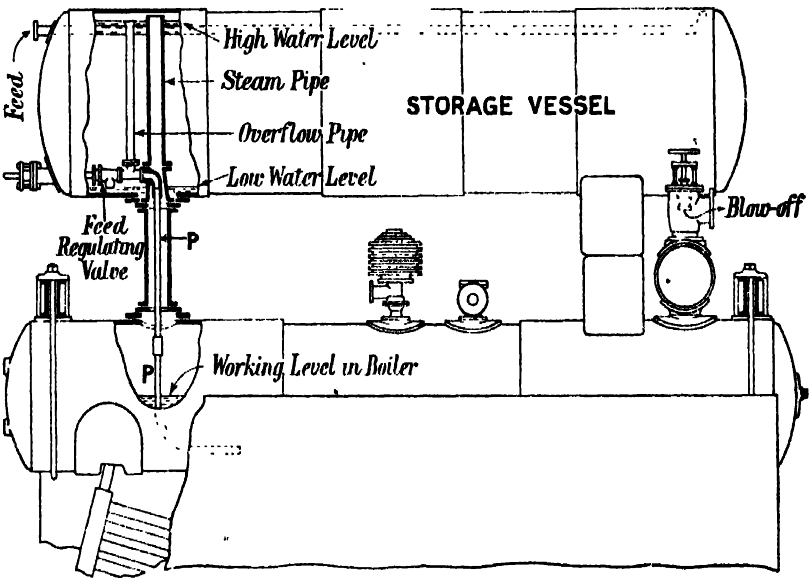


FIG. 209.—Thermal storage system.

the pipe P, its rate of flow being regulated to the evaporation in the boiler by the feed-regulating valve shown. The water in the storage vessel is heated by steam from the boiler which passes through the steam pipe shown, this pipe being always open.

CHAPTER X

NATURAL AND ARTIFICIAL DRAUGHT

146. Volume of Chimney Gases.—An examination of the equation $C + O_2 = CO_2$, which represents the combustion of carbon, shows that the volume of carbon dioxide produced is equal to the volume of oxygen used, and if the nitrogen associated with this oxygen and also the excess air be added to both sides of the equation it follows that the volume of the chimney gases from carbon fuel is equal to the volume of the air used, the volumes being measured at the same temperature.

In the combustion of hydrogen the equation $2H_2 + O_2 = 2H_2O$ shows that the volume of the steam produced is twice the volume of the oxygen used. But since the volume of steam in the products of combustion of most fuels is a small fraction of the total volume it will generally be sufficiently accurate to assume that the volume of the products of combustion or the volume of the chimney gases is equal to the volume of the air used, the volumes being measured at the same temperature. (See Exercise 1, p. 215.)

Let w = weight of air used in lb. per lb. of fuel, then the weight of the products of combustion $= w + 1$ per lb. of fuel.

Volume of products $= 12{\cdot}39\, w$ cubic feet at 0° C. per lb. of fuel.

Density of products $= \dfrac{w+1}{12{\cdot}39w}$ lb. per cubic foot at 0° C.

Let the temperature of the products or chimney gases be $t°$ C., or $t + 273 = \mathrm{T}$ absolute.

Volume of chimney gases at $t°$ C. $= 12{\cdot}39w \cdot \dfrac{\mathrm{T}}{273}$ cubic feet per lb. of fuel.

Density of chimney gases at $t°$ C. $\doteq \dfrac{w+1}{12{\cdot}39w} \cdot \dfrac{273}{\mathrm{T}}$ lb. per cubic foot.

147. Chimney Draught.—The draught produced by a chimney is due to the difference between the weight of the column of hot gas within the chimney and the weight of an equal column of cool air outside the chimney. Let H (Fig. 210) be the height of the chimney, in feet, measured from the level of the grate of the furnace. Let T be the absolute temperature of the column of gas within the chimney and let T_1 be the absolute temperature of the cool air outside. The pressure at the level of the grate due to the column of hot gas of height H, in lb. per square foot, is $\dfrac{w+1}{12{\cdot}39w} \cdot \dfrac{273}{\mathrm{T}} \cdot \mathrm{H}$.

The pressure at the grate due to a column of cool air of height H, in lb. per square foot, is $\frac{1}{12\cdot39}\cdot\frac{273}{T_1}\cdot H$. The pressure causing the draught is therefore

$$\frac{1}{12\cdot39}\,\frac{273}{T_1}\cdot H - \frac{w+1}{12\cdot39w}\cdot\frac{273}{T}\cdot H$$

$$= \frac{273H}{12\cdot39}\left(\frac{1}{T_1} - \frac{w+1}{w}\cdot\frac{1}{T}\right) = P \text{ lb. per sq. ft.}$$

Let h be the height of a column of hot gas (absolute temperature T) which would produce the pressure P. Then

$$h = P \div \frac{w+1}{12\cdot39w}\,\frac{273}{T} = \frac{273H}{12\cdot39}\left(\frac{1}{T_1} - \frac{w+1}{w}\cdot\frac{1}{T}\right) \div \frac{w+1}{12\cdot39w}\cdot\frac{273}{T}$$

Therefore $$h = H\left(\frac{w}{w+1}\cdot\frac{T}{T_1} - 1\right).$$

If h' is the height, in inches, of a column of water which will produce the pressure P, then $h' = \frac{12P}{62\cdot3} = 4\cdot244H\left(\frac{1}{T_1} - \frac{w+1}{w}\cdot\frac{1}{T}\right)$. P

The height h' would be that shown by the ordinary **U**-tube water pressure gauge.

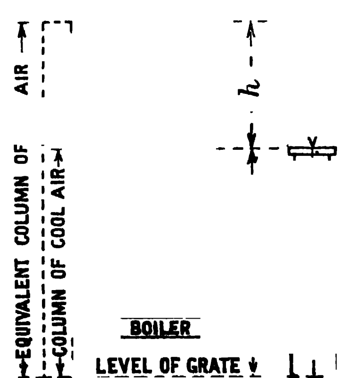


FIG. 210.

The chimney draught is most effective when the maximum weight of hot gas is discharged in a given time and it will now be shown that this occurs when the absolute temperature of the chimney gases bears a certain ratio to the absolute temperature of the outside air.

The velocity of the hot gas in the chimney is proportional to $\sqrt{h}$ and is therefore proportional to $\sqrt{\left(\frac{w}{w+1}\cdot T - T_1\right)}$.

Again, the density of the hot gas is proportional to $\frac{1}{T}$.

But the weight of hot gas discharged in a given time is proportional to the product of the velocity and density and is therefore proportional to $\frac{1}{T}\sqrt{\left(\frac{w}{w+1}\cdot T - T_1\right)}$. Differentiating this expression and equating the result to zero it is found that $\frac{1}{T}\sqrt{\left(\frac{w}{w+1}\cdot T - T_1\right)}$ has its maximum value when $T = \frac{2(w+1)}{w}\cdot T_1$. Inserting this value of T in the equation, $h = H\left(\frac{w}{w+1}\cdot\frac{T}{T_1} - 1\right)$, gives $h = H$.

If the temperature of the outside air is 10° C., then $T_1 = 283$. The temperatures of the chimney gases for most effective draught corresponding to $w = 12$, $w = 18$, and $w = 24$, when $T_1 = 283$ are given in the following table:—

w, the weight of air used in lb. per lb. of fuel	12	18	24
Temperature of chimney gases, deg. C. ab. scale	613	597	590
,, ,, ,, ,, C. ord. ,,	340	324	317
,, ,, ,, ,, F. ,, ,,	644	615	602

Since the weight of gas discharged by the chimney in a given time is proportional to $\frac{1}{T}\sqrt{\left(\frac{w}{w+1}\cdot T - T_1\right)}$ and since the maximum discharge takes place when $T = \frac{2(w+1)}{w}\cdot T_1$ it follows that the maximum discharge is proportional to $\frac{w}{2(w+1)\sqrt{T_1}}$.

Hence when T is not equal to $\frac{2(w+1)}{w}\cdot T_1$ the actual discharge expressed as a percentage of the maximum discharge is

$$\frac{\frac{100}{T}\sqrt{\left(\frac{w}{w+1}\cdot T - T_1\right)}}{\frac{w}{2(w+1)\sqrt{T_1}}} = \frac{200(w+1)\sqrt{T_1}}{wT}\sqrt{\left(\frac{w}{w+1}\cdot T - T_1\right)}$$

It is instructive to plot this against the temperature of the chimney gases as shown in Fig. 211 for $w = 12$ and $w = 24$ when the temperature of the air is 10° C., or $T_1 = 283$.

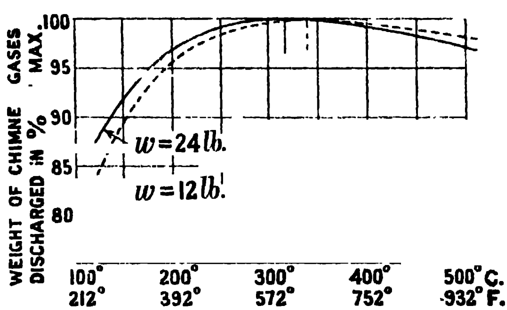


FIG. 211.

It will be seen that the temperature of the chimney gases may be lowered considerably below the temperature for maximum discharge without reducing the discharge to any considerable extent.

In addition to the symbols already used, let—

W = weight of fuel in lb. burned per square foot of grate per hour.
a = total area of grate in square feet.
A = cross sectional area of chimney at top in square feet.
u = velocity of gases through top of chimney in feet per second.

Volume of gases leaving chimney per second $= \frac{12{\cdot}39wWa}{60 \times 60}\cdot\frac{T}{273}$

Hence $u = \frac{12{\cdot}39wWa}{60 \times 60A}\cdot\frac{T}{273} = \frac{wWaT}{79322A}$

The relation between u and h the draught pressure in feet of hot gas, allowing for friction, is given by the following formula due to Peclet —

$$h = \frac{u^2}{2g}\left(1 + G + \frac{fl}{m}\right)$$

Where l = whole length of chimney and flue leading to it from the furnace, in feet.

m = hydraulic mean depth of flue and chimney = area of cross section divided by its perimeter. For a square section m is one-fourth of the side of the square, and for a circular section m is one-fourth of the diameter.

f = coefficient of friction = 0·012 for gas moving over sooty surfaces.

G = factor of resistance for the passage of the air through the grate and layer of fuel above it = 12 when W is 20 to 24.

Inserting the above values of the constants f and G,

$$h = \frac{u^2}{2g}\left(13 + \frac{0{\cdot}012l}{m}\right)$$

It has already been shown that $h = \mathrm{H}\left(\frac{w}{w+1}\cdot\frac{\mathrm{T}}{\mathrm{T}_1} - 1\right)$, also $u = \frac{w\mathrm{W}a\mathrm{T}}{79322\mathrm{A}}$.

Hence, $$\mathrm{H}\left(\frac{w}{w+1}\cdot\frac{\mathrm{T}}{\mathrm{T}_1} - 1\right) = \left(\frac{w\mathrm{W}a\mathrm{T}}{79322\mathrm{A}}\right)^2\frac{1}{2g}\left(13 + \frac{0{\cdot}012l}{m}\right)$$

a formula which may be used in designing chimneys or to find the performance of a given chimney. The formula is based on the assumption that A and m are the same for the flue and chimney throughout, which is not generally the case. When a number of boilers of the same size and type are connected to the same chimney the length l may be taken as the length of flue for one boiler from the furnace to the chimney plus the height of the chimney.

The following example illustrates the application of the foregoing formula to the determination of the performance of a chimney. Given: H = 150, l = 150 + 90 = 240, A = 0·7854 × 6^2 = 28·3, $m = \frac{6}{4}$ = 1·5, w = 20, T = 257 + 273 = 530, T_1 = 7 + 273 = 280. It is required to determine x the weight of coal burned per hour. $x = \mathrm{W}a$.

$$150\left(\frac{20}{21} \times \frac{530}{280} - 1\right) = \left(\frac{20 \times 530}{79322 \times 28{\cdot}3}\right)^2 \frac{x^2}{2 \times 32{\cdot}2}\left(13 + \frac{0{\cdot}012 \times 240}{1{\cdot}5}\right)$$

Therefore x = 4828 lb. per hour. If the rate of combustion is 20 lb. per square foot of grate per hour, then the total grate area would be 4828 ÷ 20 = 241·4, which would be that of eight Lancashire boilers each with a grate area of about 30 square feet.

148. Empirical Formulæ for Chimneys.—The formula towards the end of the preceding Art., namely,

$$\mathrm{H}\left(\frac{w}{w+1}\cdot\frac{\mathrm{T}}{\mathrm{T}_1} - 1\right) = \left(\frac{w\mathrm{W}a\mathrm{T}}{79322\mathrm{A}}\right)^2\frac{1}{2g}\left(13 + \frac{0{\cdot}012l}{m}\right)$$

may be simplified, as follows, for practical use. The factor $\left(13 + \frac{0 \cdot 012 l}{m}\right)$ may be taken equal to 16, $w = 20$, $T = 277 + 273 = 550$, and $T_1 = 10 + 273 = 283$.

Then $$H = \left(\frac{0 \cdot 075 Wa}{A}\right)^2 = 0 \cdot 0056 \left(\frac{Wa}{A}\right)^2$$

The following formula by Mr. William Kent has been considerably used in practice

$$P = 3 \cdot 33 \left(A - 0 \cdot 6 \sqrt{A}\right) \sqrt{H}$$

where P is the rated horse-power of the boilers discharging their gases into the chimney considered, it being assumed that 5 lb. of fuel is burned per horse-power per hour.

Hence $5 \times 3 \cdot 33 \left(A - 0 \cdot 6 \sqrt{A}\right) \sqrt{H} = Wa$,

and therefore $H = \left(\frac{0 \cdot 06 Wa}{A - 0 \cdot 6 \sqrt{A}}\right)^2 = 0 \cdot 0036 \left(\frac{Wa}{A - 0 \cdot 6 \sqrt{A}}\right)^2$

149. Efficiency of a Chimney.—When a chimney is used to produce a draught it is necessary to have a temperature of waste gases higher than the minimum required when the draught is produced in some other way. Suppose that the temperature of the chimney gases is 320° C. (608° F.) and that with artificial draught the minimum temperature of the waste gases as they escape into the atmosphere would be 150° C. (302° F.). Then the extra heat carried away by the waste gases due to the higher temperature required to produce the natural draught is $(320 - 150) \times 0 \cdot 24 = 40 \cdot 8$ C.H.U. per lb. of gas, 0·24 being taken as the mean specific heat of the waste gases between 150°C. and 320°C. The mechanical equivalent of this heat is $40 \cdot 8 \times 1400 = 57{,}120$ foot-lb. The draught pressure in feet of hot gas is $h = H\left(\frac{w}{w+1} \cdot \frac{T}{T_1} - 1\right)$. Let $H = 100$, $w = 20$, and $T_1 = 10 + 273 = 283$. Then $h = 99 \cdot 6$ feet. The maximum energy which this head will give to each lb. of gas is 99·6 foot-lb., and this is at the expense of an amount of heat whose mechanical equivalent is 57,120 foot-lb. The efficiency of the chimney is therefore $\frac{99 \cdot 6 \times 100}{57{,}120} = 0 \cdot 17$ per cent.

The efficiency is evidently proportional to the height of the chimney, but even with the highest chimney ever constructed (less than 500 feet) the efficiency is less than 1 per cent.

The temperature of the air being 10° C., then in the foregoing example the heat really used per lb. of waste gas to produce the draught is $(320 - 10) \times 0 \cdot 24 = 74 \cdot 4$ C.H.U., but the difference between this and the 40·8 C.H.U. found above would be carried away and lost in any case and is therefore not charged to the chimney draught.

150. Expenditure of Heat on Chimney Draught.—Using the data given in the preceding Art., for every lb. of fuel burned there are 21 lb. of waste gases and the excess heat which these gases carry away in order that the chimney draught may be produced is $(320 - 150) \times 0 \cdot 24 \times 21 = 856 \cdot 8$ C.H.U. per lb. of fuel burned. Taking the calorific

value of the fuel as 8000 C.H.U. per lb., the heat spent on draught is therefore 10·7 per cent. of the heat of the fuel in the case considered.

151. Artificial Draught.—The draught produced by a chimney is known as *natural draught* and draught produced in any other way is called *artificial draught*. The principal contrivances used for producing artificial draught are the fan and the steam jet. The use of the steam jet for draught production is mainly confined to the steam locomotive.

The first point to notice about artificial draught is that the comparatively high temperature of the waste gases necessary for natural draught is not required when artificial draught is used. With artificial draught the reduction of the temperature of the waste gases is only limited by the requirements of the apparatus such as feed-water heaters or economizers used for utilizing the heat in the gases leaving the boiler.

In practice there is no serious difficulty in reducing the temperature of the waste gases to, say, 100° C. (212° F.) by giving the heat to the feed-water. If the waste gases leaving the boiler have a temperature of say 325° C. (617° F.) and if the weight of the waste gases is 20 lb. per lb. of fuel then a reduction of temperature to 100° C. means a saving of heat amounting to $(325 - 100) \times 0{\cdot}24 \times 20 = 1080$ C.H.U., which is 13·5 per cent. of the heat of a fuel having a calorific value of 8000 C.H.U. Of course against this saving must be placed the cost of the artificial draught and feed-heating plant less the cost of the chimney for natural draught. By cost is here meant maintenance, depreciation, and interest on the first cost of the plant referred to.

When artificial draught was first introduced for boilers it was not simply as an alternative to natural draught, but because a higher draught pressure was required than could be conveniently or economically produced by a chimney alone. An increase of draught pressure is attended by a higher rate of combustion and a higher furnace temperature, and a less quantity of surplus air is sufficient for complete combustion. A higher rate of combustion and a higher furnace temperature means more heat produced and more steam generated in a given time. It follows therefore that with higher draught pressure a smaller boiler will develop the same power. Again, less surplus air means less weight of waste gases per lb. of fuel and therefore less loss of heat in these gases.

One great advantage which artificial draught has over natural draught is that it may easily be regulated to suit the immediate requirements of the furnace. A sudden demand for more steam may be met by at once increasing the draught pressure and the supply of air necessary for an increased rate of combustion of fuel.

Artificial draught is to a large extent independent of climatic conditions which often seriously affect natural draught.

With the higher intensity of draught which may be produced artificially inferior qualities of fuel may be economically burned.

152. Fan Draught.—The mechanical efficiency of a fan, measured by the ratio of the energy put into the air to the work to drive the fan, may be taken at not less than 50 per cent. The fan will generally get its power directly or indirectly from a steam engine and boiler, the combined efficiency of which may be taken at 10 per cent. Hence the

resultant efficiency of the fan may be taken at 5 per cent. But it has been shown in Art. 149 that the efficiency of natural draught is generally considerably less than 1 per cent. A fan is therefore a much more efficient instrument for producing draught than a chimney. Also, the first cost of a fan and a motor to drive it will generally be considerably less than that of a chimney to produce the same draught. In small boiler installations, however, where high efficiency is not so important and where a comparatively low chimney is sufficient, natural draught is to be preferred.

There are two principal ways of applying fan draught. In one, known as *forced draught*, the air passes through the fan before entering the furnace, and in the other, known as *induced draught*, the fan is placed near the base of the chimney and draws the air through the furnace. Since all the hot waste gases must pass through an induced draught fan, such a fan has a much greater volume to deal with than a forced draught fan for the same boilers. If T_1 is the absolute temperature of the air, and if T is the absolute temperature of the gases passing through an induced draught fan, then the volume of gas to be dealt with by the induced draught fan is greater than the volume of air to be dealt with by the forced draught fan in the ratio of T to T_1. When $T = 170^\circ$ C. $+ 273 = 443$, and $T_1 = 10^\circ$ C. $+ 273 = 283$ this ratio is is 1·57 to 1.

For the same intensity of draught and the same weight of air or gas per minute, the induced draught fan will be larger and require considerably more power than the forced draught fan. The induced draught fan must be specially designed to withstand the comparatively high temperature of the gases passing through it.

There are two systems of forced draught, the *closed stokehold* system, and the *closed ashpit* system. In the closed stokehold system, chiefly used in the navy, the stokehold or room containing the boilers is made airtight and the fan forces the air into the stokehold so that the atmosphere in which the stokers work is under a pressure greater than that of the outside air. Communication between the stokehold and the outside is by means of double doors with a space between them forming an air lock.

In the closed ashpit system the front of the ashpit is closed and the delivery pipe from the fan leads into ducts which conduct the air to the ashpits of the various furnaces; part of the air is delivered above the grate but the greater portion passes from the ashpit through the grate.

Comparing induced draught and the two systems of forced draught, induced draught is like natural draught in that the pressure within the furnace and boiler flues is less than that of the atmosphere and any leakage in the flues will be from the outside into the flues, increasing the volume of gas to be dealt with by the fan and increasing the heat carried up the chimney.

In both systems of forced draught the pressure within the furnace and boiler flues is greater than that of the atmosphere and any leakage in the flues will be from the flues to the outside. With the closed ashpit system, the pressure within the furnace being greater than that in the stokehold, the forced air supply must be cut off when the furnace

door is opened, otherwise the flames will be driven out with probable serious consequences to the stoker. It is best to arrange that the opening of the furnace door automatically shuts off the forced air supply.

153. **Steam Jet Draught.**—A jet of steam issuing from a nozzle placed in a pipe or chimney will drag with it the surrounding air or gas, producing a partial vacuum in its neighbourhood. The amount of the vacuum or the intensity of the draught depends on the pressure of the steam, the diameter of the nozzle, and the diameter of the pipe or chimney. The axis of the nozzle should be in line with the axis of the pipe or chimney. Instead of a single jet there may be a considerable number of jets proceeding from small holes in a hollow ring, in which case the axis of the ring should be in line with the axis of the pipe or chimney in which the draught is to be produced and the axes of the jets should be parallel to the axis of the ring.

When steam for the steam jet is taken direct from the boiler the amount of steam to produce the necessary draught will not be less than 5 per cent. of the whole steam produced by the boiler and will more generally be from 10 to 15 per cent. Now since the steam required by an engine to drive a fan to produce the draught is only from 1 to 2 per cent. of the steam produced by the boiler it is seen that steam jet draught is much more costly in steam consumption than fan draught. A steam nozzle is, however, simple, cheap to install, and does not readily get out of order.

The steam jet may be placed in the chimney or in the ashpit. An objection to placing the steam jet in the ashpit is that the steam, which does not support combustion, carries with it from the furnace a considerable amount of superheat into the chimney, the specific heat of the steam being about double that of the other waste gases. It is however found that with inferior qualities of fuel, which, when burned form a considerable amount of clinker which clogs the grate, it is advantageous to have a steam jet in the ashpit. In this case the decomposition of the steam which takes place when it passes through the red-hot fuel has a chilling effect which prevents the formation of clinker. The heat lost by this decomposition is recovered in the burning of the liberated hydrogen above the fire.

When the steam generated in a boiler is used in a non-condensing engine the boiler draught may be produced at little extra cost by directing the exhaust steam from the engine through a blast pipe and nozzle in the chimney, as is always done in steam locomotives. In this case the greater the power required of the engine the greater is the amount of steam to be delivered by the boiler and the greater is the amount of the exhaust steam, and consequently the greater the draught produced. The draught is therefore automatically adjusted to suit the requirements of the boiler.

154. **Heating Air Supply.**—In fan draught systems it is a common practice to heat the air on its way to the furnace by causing it to pass through or over a nest of tubes over which or through which the waste gases are made to pass. In this way the air before entering the furnaces may be raised in temperature to 120° C. (248° F.) or even to 175° C. (347° F.), depending on the heat available in the waste gases.

If t_1 is the temperature of the atmosphere, t_2 the temperature to which the air is raised by the heat from the waste gases, and w the weight of air used per lb. of fuel, then the direct saving of heat per lb. of fuel is $0{\cdot}233\, w(t_2 - t_1)$, where 0·233 is the mean specific heat of the air. This is however not the whole of the saving due to the heating of the air supply. There is, in addition, the advantage of a higher temperature of combustion in the furnace.

155. Intensity of Draught and Rate of Combustion.—Theory indicates that the velocity of the air through the grate and the fire above it is proportional to the square root of the difference of pressure under and over the fire. But the rate of combustion of the fuel is approximately proportional to the rate at which the air is supplied for its combustion. Hence, if W is the rate of combustion in lb. of fuel per square foot of grate per hour, h_1 and h_2 the pressures under and over the fire, in inches of water, then approximately $W = c\sqrt{h_1 - h_2}$, where c is a constant. In tests made by Mr. E. Lechner[1] with a locomotive type of boiler on a torpedo boat, c varied from about 72 to 86 and had an average value of about 80. In these tests the closed stokehold pressure varied from 1·97 to 6·8 inches of water.

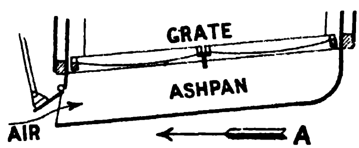


FIG. 212.

With natural draught from a chimney of moderate height, producing a vacuum at its base of from $\frac{1}{4}$ to $\frac{1}{2}$ inch of water, the rate of combustion is from 15 to 25 lb. per square foot of grate per hour. With a high chimney producing a vacuum at its base of about $\frac{3}{4}$ inch of water the rate may reach 30 lb. per square foot of grate per hour.

With moderate forced draught producing a pressure in the ashpit of $\frac{1}{2}$ to 1 inch of water in addition to a chimney draught of about $\frac{1}{4}$ inch of water the rate of combustion is from 25 to 35 lb. per square foot per hour. In water-tube boilers a rate of 50 to 60 lb. per square foot per hour is obtained with a pressure in the stokehold of 2 to 3 inches of water.

The highest rates of combustion occur in locomotive boilers. The ordinary rate in these boilers is from 60 to 90 lb. per square foot of grate per hour, and it sometimes reaches 140 lb.

The vacuum in the smoke-box of a locomotive when running varies from 3 to 8 inches of water, and in the fire-box near the tube plate from 1 to 3 inches. In the ashpan under the grate the air pressure is generally from 0·2 to 0·8 inch of water. The pressure in the ashpan is due to the velocity of the locomotive through the air as shown in Fig. 212, where the arrow A shows the direction of the motion of the train.

If V is the speed of the train, in miles per hour, in relation to that of the air in front of the ashpan, that is, V is the velocity of the air into the ashpan, then, the theoretical maximum pressure of the air in the ashpan is about $\left(\frac{V}{45}\right)^2$ in inches of water, but the actual pressure is generally considerably less than this.

[1] *Proceedings of the American Society of Naval Architects*, 1891.

A locomotive boiler is only able to take advantage of the high rate of combustion occurring in its fire-box because of the oscillations and vibrations to which it is subjected when the locomotive is running on the rails. These oscillations and vibrations facilitate the circulation of the water and the escape of the steam from the water. The same boiler used as a stationary boiler on a rigid foundation cannot utilize anything like so high a rate of combustion as occurs in the running locomotive

Exercises X

1 Justify the statement made in Art. 146, p. 206, namely, that the volume of the products of combustion is approximately equal to the volume of air used, by calculating the volume of the products per lb. of a coal whose percentage composition is: C, 87; H, 5; O, 6; and ash, 2; (*a*) when the air used is the minimum for complete combustion, (*b*) when the air used is one and a half times the minimum, and (c) when the air used is twice the minimum. Also calculate the volume of the air used in each case. Assume that the air and products have a temperature of 300° C. Given that the specific volumes of air, oxygen, and nitrogen are, 12·39, 11·21, and 12·80 cubic feet per lb. respectively at 0° C.

2. Given: Temperature of chimney gases = 250° C, temperature of air = 10° C., weight of air used per lb. of fuel = 20 lb. Determine the height of chimney necessary to produce a draught represented by a column of water 0·6 inch high.

3. A chimney has a height of 100 feet. The temperature of the air is 12° C. Find the draught in inches of water when the temperature of the chimney gases is such as to cause the weight of these gases discharged in a given time to be a maximum.

4. Three boilers have a combined grate area of 100 square feet. Use the formula at the end of Art. 147 to calculate the height of chimney for these boilers, having given: Diameter of chimney, 4 feet, $l = 90 + H$, $w = 20$, $T = 267 + 273 = 540$, $T_1 = 12 + 273 = 285$, and $W = 18$.

5. Find the answer to the preceding exercise by means of Kent's formula given in Art. 148, p. 210.

6. Use Kent's formula (Art. 148) to calculate the diameter of a chimney for a group of boilers burning 3500 lb. of coal per hour, the height of the chimney being 145 feet.

7. Apart from the question of draught the minimum temperature of the waste gases leaving a certain boiler installation is 130° C. A natural draught is produced by means of a chimney 150 feet high, the gases within the chimney having a temperature of 310° C. The air supply to the furnaces is 19 lb. per lb. of fuel burned, the temperature of the air is 7° C., and the mean specific heat of the waste gases is 0·24. Calculate the efficiency of the chimney as an instrument for moving the air and products of combustion through the furnaces and flues.

If the calorific value of the fuel is 7800 C.H.U. per lb. express the extra expenditure of heat on the production of the chimney draught as a percentage of the heat of the fuel.

8. A given weight of air has to be supplied per minute to a given boiler furnace by means of a fan. Show clearly, that if induced draught is used considerably more power will be required to drive the fan than if forced draught is employed, notwithstanding the fact that the work to be done in moving the air and waste gases may be about the same in the two cases and that the efficiencies of the fans may be equal.

[Note that the pressure energy of 1 lb. weight of a gas is P_1/w_1 foot-lb., where P_1 is the static pressure of the gas in lb. per square foot, and w_1 is the weight of 1 cubic foot of the gas in lb. Also the kinetic energy of 1 lb. weight of a gas is $v_1^2/2g$ where v_1 is its velocity in feet per second.]

9. Given: Rate of combustion, 25 lb. per square foot of grate per hour; weight of air supplied, 20 lb. per lb. of fuel burned; area of openings between fire bars, 40 per cent. of area of grate; volume of 1 lb. of air at 0° C., 12·39 cubic feet. Calculate the velocity of the air between the fire bars, in feet per second,

when 90 per cent. of the air supply passes through the grate and the temperature of the air between the fire bars is 500° C.

10. The calorific value of a fuel is 7900 C.H.U. and the weight of air supplied for its combustion is 17 lb. per lb. of fuel. The air is heated by the waste gases before entering the furnace from 9° C. to 140° C. Taking the mean specific heat of the air as 0·238, calculate the direct saving of heat due to pre-heating the air supply, expressing the saving as a percentage of the heat of the fuel.

11. The following particulars relate to a test of a large American freight locomotive:—Grate area, 31 square feet; weight of coal burned per hour 2413 lb.; weight of engine and train, 1670 tons; average speed, 16 miles per hour, Calculate, (a) the rate of combustion in lb. per square foot of grate per hour. (b) the weight of coal burned per ton mile.

12. The pressure in the ashpan of a locomotive boiler is 0·9 inch and the vacuum in the smoke-box is 5·2 inches of water. What is the effective draught between the ashpan and the smoke-box in inches of water and in lb. per square foot?

13. Assuming that the formula $W = 80\sqrt{h_1 - h_2}$ gives the rate of combustion W in lb. per square foot of grate per hour when h_1 and h_2 are the pressures in inches of water below and above the fire respectively, plot W and $h_1 - h_2$ for values of W from 10 to 100.

CHAPTER XI

PERFORMANCE OF STEAM BOILERS

156. Equivalent Evaporation — Factor of Evaporation. — The purpose of a steam boiler is to evaporate water by heat obtained by the combustion of fuel and the amount of water evaporated is therefore one of the quantities to be considered in dealing with the performance of a steam boiler. The evaporation is generally stated in lb. per lb. of fuel, or in lb. per square foot of heating surface per hour, or the total weight of water evaporated may be stated. But the amount of water evaporated by a boiler is not a sufficiently definite measure of its performance because under different conditions as to temperature of feed water, and temperature and dryness of steam produced, a given evaporation will represent different amounts of heat utilized by the boiler.

For the purpose of comparison it is therefore necessary that the water be supposed to be evaporated under standard conditions. The standard conditions which have been adopted are: temperature of feed-water and temperature of steam 100° C. (212° F.), and steam dry and saturated. Under standard conditions therefore the evaporation of 1 lb. of water represents the utilization of L_0 units of heat where L_0 is the latent heat of steam at 100° C. (212° F.) and has the value 539 C.H.U. (970 B.Th.U.).

If under actual conditions—

h_1 = heat of feed water per lb.
h = sensible heat of steam per lb.   } as given in saturated steam table
H = total heat of steam per lb.
L_0 = latent heat of steam at 100° C. (212°F.) per lb.
x = dryness fraction of steam
W = weight of steam produced

then, heat to produce 1 lb. of steam under actual conditions $= (H - h)x + h - h_1$, and the heat to produce W lb. of steam under actual conditions $= W\{(H - h)x + h - h_1\}$. This amount of heat, if applied to produce steam under standard conditions, would evaporate W_1 lb. of water, and

$$W_1 = \frac{W\{(H - h)x + h - h_1\}}{L_0}$$

W_1 is called the *equivalent evaporation* and the expression $\frac{(H - h)x + h - h_1}{L_0}$ is called the *factor of equivalent evaporation* or

factor of evaporation. (Equivalent evaporation is often spoken of as *evaporation from and at* 100° C. (212° F.).)

If under actual conditions the steam is dry and saturated, then $x = 1$, and the factor of evaporation becomes $\frac{H - h_1}{L_0}$. In practice x is generally 0·98 or 0·99.

Generally it is sufficiently accurate to take the sensible heat per lb. as the degrees of temperature above 0° C. (32° F.).

If under the actual conditions there is a superheater which is an integral part of the boiler, and if the steam has t degrees of superheat, then the factor of evaporation is $\frac{H - h_1 + k_p t}{L_0}$, where k_p is the mean specific heat of the steam over the range $t°$.

157. Evaporation per Pound of Fuel.—If Q is the calorific value of 1 lb. of fuel, then the theoretical maximum evaporation is Q/L_0 lb. from and at 100° C. (212° F.). The actual equivalent evaporation in ordinary steam boilers varies from 50 to 85 per cent. of the theoretical maximum and is generally from 65 to 75 per cent.

158. Evaporation per Square Foot of Heating Surface.—The equivalent evaporation in lb. per square foot of heating surface per hour in ordinary steam boilers working under ordinary conditions may be taken as follows:—

Cornish and Lancashire boilers .	3 to 10, average about 5
Marine boilers (Scotch type) . .	3 to 10, average about 6
Locomotive boilers	5 to 15, average about 11
Water-tube boilers	2·5 to 6, average about 4

The evaporation per square foot of heating surface per hour is obtained by dividing the total weight of water evaporated per hour by the total area of the heating surface, a result which is the *mean* evaporation per square foot. The evaporation is different at different parts of the heating surface. For example, in a locomotive boiler the heating surface of the fire-box is much more effective than that of the tubes.

In the Bonecourt surface combustion boiler (p. 142) an equivalent evaporation as high as 35 lb. per square foot of boiler heating surface has been obtained.

159. Power of a Boiler—Boiler Horse-Power.—A definite and fairly common way of stating the *power* of a steam boiler is to give the weight of water which it is capable of evaporating from and at 100° C. (212° F.) in an hour. The power of any particular boiler must of course depend on the quality of the fuel used and the rate of its combustion and also on the efficiency of the boiler in utilizing the heat of combustion of the fuel.

A committee of the American Society of Mechanical Engineers has recommended that the standard unit of power for steam boilers be an evaporation of 34·5 lb. of water from and at 100° C. (212° F.) in an hour. This unit of boiler power is called a *boiler horse-power* and it is used to a considerable extent, especially in America.

The horse-power of a boiler must not be confused with the horse-power of the engine which uses the steam from the boiler. The

horse-power of the engine will generally be greater than the horse-power of its boiler as measured by the above rule. For example, a triple-expansion condensing engine taking dry saturated steam at a pressure of 150 lb. per square inch absolute, used 13 lb. of steam per indicated horse-power per hour. The feed water had a temperature of 65° C. (149° F.). From these data and the steam tables the equivalent evaporation of the boiler would be 14·4 lb. per indicated horse-power of the engine per hour. The indicated horse-power of the engine would therefore be $34·5 \div 14·4 = 2·4$ times the boiler horse-power.

160. Efficiency of a Boiler.—Knowing the weight of steam produced by a boiler plant in a given time, the temperature of the feed water, and the pressure and the dryness or superheat of the steam, the amount of heat (Q_1) utilized by the boiler plant may be computed. Knowing also the weight of fuel burned in the same time and its calorific value, the maximum possible amount of heat (Q) due to the combustion of this fuel may be calculated. The ratio of the heat (Q_1) utilized to the heat (Q) supplied in the same time is the *efficiency of the boiler plant.*

The efficiency of a boiler plant may also be computed from the heat utilized per lb. of fuel and the calorific value of 1 lb. of the fuel.

The essential part of a boiler plant is the boiler, but there may also be a feed water heater or economizer and there may be a superheater. Each of these elements of a boiler plant has its own efficiency measured by the ratio of the heat which it utilizes to the heat supplied to it.

EXAMPLE.—The following observations and deductions are taken from a report of a trial of a boiler plant consisting of six Lancashire boilers and an economizer.

Calorific value of coal, per lb.	7150 C.H.U.
Weight of feed water evaporated per lb. of dry coal	9·1 lb.
Equivalent evaporation from and at 100° C. per lb. of dry coal	9·6 lb.
Temperature of feed to economizer . .	12° C.
Temperature of feed to boiler	105° C.
Temperature of air	13° C.
Temperature of flue gases entering economizer	370° C.
Weight of flue gases per lb. of dry coal .	18·2 lb.
Mean specific heat of flue gases . . .	0·25.

From the above it is required to find the efficiency of the boilers alone, the efficiency of the economizer alone, and the efficiency of the whole plant.

Heat utilized by boilers per lb. of dry coal $= 9·6 \times 539 = 5174$ C.H.U.

Heat of combustion of 1 lb. of dry coal = 7150 C.H.U.

$$\textit{Efficiency of boilers} = \frac{5174}{7150} = 0·72 \text{ or } 72 \text{ per cent.}$$

Heat utilized by economizer per lb. of dry coal

$$= 9 \cdot 1 \times (105 - 12) = 846 \text{ C.H.U.}$$

Heat in flue gases entering economizer, reckoned above temperature of air $= 18 \cdot 2 \times 0 \cdot 25 \times (370 - 13) = 1624$ C.H.U.

$$\textit{Efficiency of economizer} = \frac{846}{1624} = 0 \cdot 52 \text{ or } 52 \text{ per cent.}$$

Heat utilized by boilers and economizer per lb. of dry coal $= 5174 + 846 = 6020$ C.H.U.

$$\textit{Efficiency of whole plant} = \frac{6020}{7150} = 0 \cdot 84 \text{ or } 84 \text{ per cent.}$$

161. Heat Losses in Boiler Plant.—That portion of the heat of combustion of the fuel which is not utilized in the production of the steam from the feed water is reckoned as lost heat. The sources of loss are: (1) Hot chimney gases. (2) Incomplete combustion of carbon, that is, the burning of carbon to carbon monoxide (CO) instead of to carbon dioxide (CO_2). (3) Unburned fuel which may drop into the ashpit or be carried away with the products of combustion. (4) Heating, evaporating, and superheating moisture in the fuel. (5) External radiation.

These various heat losses are considered in detail in the next five Arts.

162. Heat Loss in Chimney Gases.—The heat carried away in the chimney gases may be divided into two parts, (*a*) the heat in the products of combustion, and (*b*) the heat in the excess air.

Let t = temperature of chimney gases.

t_1 = temperature of air.

k_p = mean specific heat of the mixture of gases between the temperature t_1 and t.

w = weight of chimney gases in lb. per lb. of dry fuel.

Q = calorific value of 1 lb. of dry fuel.

The heat carried away by the chimney gases per lb. of dry fuel is $wk_p(t - t_1)$ heat units and this is $\dfrac{100wk_p(t - t_1)}{Q}$ per cent. of heat of fuel.

The mean specific heat k_p depends on the composition of the gases and it also varies with the range of temperature. k_p is generally taken at about 0·25 and this is about right *if the latent heat in the steam in the chimney gases is neglected.*

The following example illustrates how the heat carried away in the chimney gases may be calculated without assuming the value of k_p for the mixture of gases.

Percentage composition of dry fuel: C, 88·4; H, 3·7; O, 2·5; ash, 5·4.

Higher calorific value of dry fuel, 8300 C.H.U. (14,940 B.Th.U.) per lb.

Minimum weight of air for complete combustion of 1 lb. of dry fuel, 11·42 lb.

Products of combustion per lb. of dry fuel (no excess air): CO_2, 3·24 lb.; steam, 0·33 lb ; N, 8·8 lb. Total 12·37 lb.

Temperature of chimney gases, 310° C. (590° F.).

Temperature of air, 10° C. (50° F.).

The mean specific heats of the constituent gases from 10° C. to 100° C. are found by the formulæ in Art. 28, p. 26, to be: CO_2, 0·223; N, 0·241; air, 0·236. For superheated steam at atmospheric pressure the mean specific heat may be taken at 0·48.

The heat in the products of combustion per lb. of dry fuel (no excess air) is made up as follows :—

CO_2	$3{\cdot}24 \times (310 - 10) \times 0{\cdot}223 =$	217 C.H.U.
N	$8{\cdot}8 \times (310 - 10) \times 0{\cdot}241 =$	636 ,,
Steam	$0{\cdot}33 \times \{(100 - 10) + 539 + 0{\cdot}48 \times (310 - 100)\} =$	241 ,,
	Total . .	1094 ,,

This total is $\frac{1094 \times 100}{8300} = 13{\cdot}2$ per cent. of the heat of the fuel.

If the excess air is 50 per cent. or 5·71 lb. per lb. of dry fuel, the heat carried away by this air is $5{\cdot}71 \times (310 - 10) \times 0{\cdot}236 = 404$ C.H.U. which is $\frac{404 \times 100}{8300} = 4{\cdot}9$ per cent. of the heat of the fuel.

The heat carried away in the chimney gases when there is 50 per cent. of excess air is therefore $1094 + 404 = 1498$ C.H.U. per lb. of dry fuel, which is $\frac{1498 \times 100}{8300} = 18$ per cent. of the heat of the fuel.

If the excess air is 100 per cent. or 11·42 lb. per lb. of dry fuel, the heat carried away by this air is $11{\cdot}42 \times (310 - 10) \times 0{\cdot}236 = 809$ C.H.U. which is $\frac{809 \times 100}{8300} = 9{\cdot}7$ per cent. of the heat of the fuel.

The heat carried away in the chimney gases when there is 100 per cent. of excess air is therefore $1094 + 809 = 1903$ C.H.U. per lb. of dry fuel, which is $\frac{1903 \times 100}{8300} = 22{\cdot}9$ per cent. of the heat of the fuel.

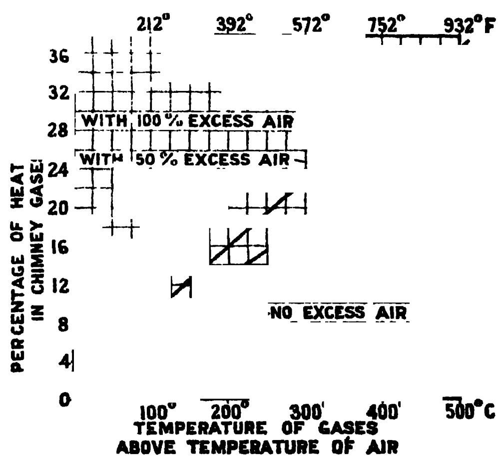


FIG. 213.

In like manner the heat carried away by the chimney gases at other temperatures may be calculated and the results plotted as shown in Fig. 213. It will be found that the graphs are practically straight lines.

If the lower calorific value of the fuel is taken, then the latent heat of the steam, 539 C.H.U. per lb., in the chimney gases must be neglected.

163. Heat Lost through Incomplete Combustion of Carbon.—The complete combustion of carbon is represented by the equation $C + O_2 = CO_2$, from which it follows that 1 lb. of carbon combines with 2·67 lb. of oxygen to form 3·67 lb. of carbon dioxide, and the result of this combination is the production of heat of the amount 8100 C.H.U.

The incomplete combustion of carbon is represented by the equation $C + O = CO$, from which it follows that 1 lb. of carbon combines with 1·33 lb. of oxygen to form 2·33 lb. of carbon monoxide, and the result of this combination is the production of heat of the amount 2416 C.H.U.

Every pound of carbon burned to CO instead of to CO_2 therefore represents a heat loss equal to 8100 − 2416 = 5684 C.H.U.

Let c_1 and c_2 be the relative volumes of CO and CO_2 in the chimney gases. The relative weights of these gases are $28c_1$ and $44c_2$. The relative weights of carbon in $28c_1$ and $44c_2$ of CO and CO_2 respectively are $\frac{12}{28} \times 28c_1$ and $\frac{12}{44} \times 44c_2$. Hence the relative weights of carbon in volumes c_1 and c_2 of CO and CO_2 respectively are the same as the relative volumes c_1 and c_2. Therefore

$$\frac{\text{weight of carbon incompletely burned}}{\text{weight of carbon burned to CO and } CO_2} = \frac{c_1}{c_1 + c_2}$$

If the fuel contains c per cent. of carbon and the whole of the carbon is burned to CO_2 and CO, then the heat lost per lb. of fuel due to the incomplete combustion of the carbon is

$$\frac{c}{100} \times \frac{c_1}{c_1 + c_2} \times 5684 \text{ C.H.U.}$$

With ordinary rates of firing (not exceeding 30 lb. per square foot of grate per hour) and careful stoking the loss of heat due to the formation of CO instead of CO_2 should be less than 1 per cent. of the heat of the fuel.

EXAMPLE.—The flue gas analysis gave CO_2, 14·7 per cent. and CO, 0·25 per cent. by volume. The fuel analysis gave carbon 87·6 per cent by weight. The calorific value of the fuel was 8300 C.H.U. per lb.

Assuming that all the carbon in the fuel was burned to CO_2 and CO, the heat lost per lb. of fuel due to the formation of CO instead of CO_2 was $0{\cdot}876 \times \frac{0{\cdot}25}{0{\cdot}25 + 14{\cdot}7} \times 5684 = 83$ C.H.U. and this is equal to $\frac{83 \times 100}{8300} = 1$ per cent. of the heat of the fuel.

164. Heat Lost through Unburned Fuel.—With coal and other solid fuels some of the fuel may drop through between the firebars before it is completely burned, a part may also be carried away into the flues and be deposited as cinders. In the case of locomotives working at full power, the draught being then intense and the rate of combustion high, the loss of heat due to unburned fuel carried into

the smoke-box in the form of cinders and sparks may amount to 40 per cent. of the heat of the fuel fired.

With ordinary rates of combustion in stationary and marine boilers the heat lost through unburned fuel is generally from 1 to 4 per cent.

The unburned fuel may be assumed to be composed of carbon and ash.

In a boiler trial the amount of unburned carbon may be determined by collecting and weighing all the cinders and ashes and deducting the weight of ash corresponding to the weight of dry fuel fired as shown by the analysis of the fuel. For example, if for every 100 lb. of dry coal fired the weight of cinders and ashes is 9 lb., and if the ash in the dry coal as shown by the analysis is 4 per cent., then the unburned carbon in this case would be 5 per cent. of the dry coal fired.

The amount of carbon in the ashes and clinker may also be determined by chemical analysis.

The first method described above is reliable if *all* the ash is collected, but in boiler trials a considerable quantity of ash may be carried into the flues and not be collected. For example, in a certain boiler trial the ash in 100 lb. of dried fuel was, by analysis, 4 lb., and for every 100 lb. of dried fuel fired the ashes and cinders collected weighed 5 lb. If this contained all the ash the weight of unburned carbon would be 1 lb., but actual analysis of the ashes and cinders gave the carbon as 2 lb., showing that 1 lb. of ash per 100 lb. of dried fuel fired had not been collected.

Having determined the weight of unburned carbon, this weight multiplied by its calorific value is the heat lost through unburned fuel.

165. Heat Lost through Moisture in Fuel.—Any moisture in the fuel when fired has first to be heated from the temperature of the air to the boiling temperature of water at atmospheric pressure, it has then to be evaporated and finally superheated to the temperature of the chimney gases, and all the heat so used is lost. In estimating the heat used in superheating the steam the specific heat of the steam may be taken as 0·48.

One per cent. of moisture in the fuel causes a heat loss of about 0·1 per cent.

166. Heat Lost through External Radiation.—The parts of the boiler and the flue walls exposed to the external air radiate heat which is lost. The amount of this loss is difficult to determine accurately, but it is frequently assumed to be about 5 per cent. of the heat of the fuel fired.

One method of determining approximately the heat lost through external radiation is to get up steam to the working pressure, and then find out the weight of fuel per hour which must be fired to keep up that pressure while the stop valve is closed. Part of the heat of the fuel fired in such a test, after steam is up to the working pressure, goes away in the chimney gases and the remainder goes to the boiler to make up for the loss due to external radiation. The loss by radiation under working conditions will however probably be greater than when standing with steam up on account of the more rapid combustion and consequent higher temperatures in the former case.

167. Minor Heat Losses.—In addition to the heat losses which have been considered there are minor losses not usually measured, such as heat lost in the hot ashes, heat lost in superheating the moisture in the air supply, and the loss due to soot in black smoke.

168. Summary of Heat Balances in Boiler Performances.—An examination of the results of a large number of boiler trials leads to the results summarized in the following table, abnormal results being neglected.

	Range per cent.	Average per cent.
Heat transferred to water and steam. (Thermal efficiency of boiler)	60 to 81	
Heat carried away in chimney gases	13 to 28	19
Heat lost by external radiation	3 to 8	5
Other heat losses	—	4
Total heat of fuel	100	100

169. Heat Balances in Locomotive Boiler Trials.—The great variation in the heat balances of locomotive boilers due to variation in the rate of firing is shown graphically in Fig. 214, reproduced from a paper by Mr. Lawford H. Fry in the *Proceedings of the Institution of Mechanical Engineers*, 1908.

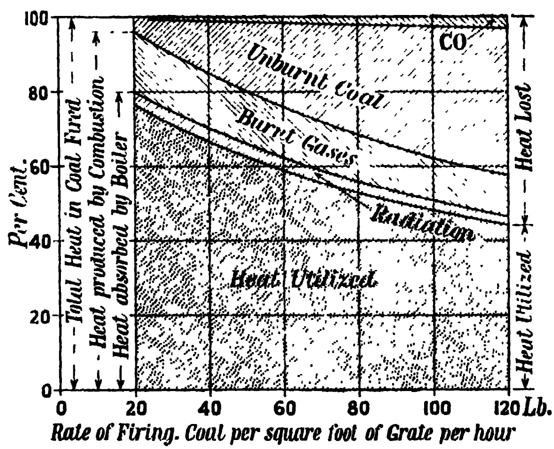


FIG. 214.

The rapid increase in the loss of heat due to unburned fuel (cinders and sparks) with the increase in the rate of firing should be specially noticed.

170. Recording Boiler Performances—The various observations to be made and the results deduced from them should be tabulated systematically. The form of tabulation most generally adopted is that recommended by a Committee of the Institution of Civil Engineers.[1] This form is shown in the following pages applied to a boiler trial made by Mr. Edward G. Hiller and reported in the *Proceedings of the Institution of Mechanical Engineers*, 1904.

[1] *Proceedings of the Institution of Civil Engineers*, vol. cl. and vol. cxcv.

General Description and Dimensions of Boiler House Plant.

Type of Boilers: Two Lancashire boilers, each with Musgrave and Dixon superheater.

Pressure for which designed: 160 lb. per square inch above atmospheric pressure.

Test made at an output of about 10,900 lb. per hour at a pressure of about 155 lb. per square inch.

Object of trial: To determine the evaporative efficiency of boilers.

General Description of Boilers and Leading Dimensions.

Ref. No.

1. Two Lancashire boilers, 8 feet diameter by 30 feet long.
   Grate area, 38 square feet each boiler.
   Water-heating surface, 920 square feet each boiler.
   Ratio of water-heating surface to grate area, 24·21 to 1.
   Superheating surface, 217 feet by 2 feet = 434 square feet.
2. Method of starting and stopping the test: The fires were cleaned, the clinkers and ashes removed, and test started with thin clear fires. The same procedure was carried out immediately before the test finished, and at the finish the fires were in same condition as at start.
3. Method of stoking: Hand firing.
4. Production of draught: By chimney.
5. Chimney: Height about 140 feet; area at bottom, 14·6 square feet; area at top, 13·25 square feet.
6. Total grate area (excluding dead plate), 76 square feet. (Two boilers.)
7. Grate area occupied by air spaces between bars —— square feet (not given).
8. Total effective heating surface of two boilers and two superheaters, 2274 square feet.
9. Capacity of water space, 2322 cubic feet. (Two boilers.)
10. Capacity of steam space, 694 cubic feet. (Two boilers.)
11. Area of water surface in boilers, 431 square feet. (Two boilers.)

Particulars of Observations.

12. Duration of trial . . . . hours 4

Fuel.

13. Description: North Wales. (Point of Ayr.)
14. Fired per hour . . . . lb. 1456
15. Analysis of coal (see footnote).[1]

[1] Note.—The actual coal used at the test was not analysed. The fuel analysis, which has been assumed for purposes of calculation, is based on the calorific determination, and is as follows: H 4·5, C 77·7, O, N, S, etc., 12·83, and ash 4·97 per cent.

A subsequent analysis of coal from the same pit gave: H 5·1, C 80·9, O, N, S., etc., 12·19, and ash 1·81 per cent. Calorific value by bomb calorimeter 14,771 B.Th.U

Ref. No.			
16.	Moisture in fuel as fired	per lb.	0·046
17.	Calorific value of dried fuel by bomb calorimeter	B.Th.U. per lb.	14,040
18.	Carbon value of 1 lb. dried fuel		0·938

Ash and Clinker.

19.	Total per hour	lb.	111·5
20.	Carbonaceous matter in ash per hour	lb.	46

Flue Gases.

21. Analysis of dry flue gases—

		By vol.	By weight.
Carbon dioxide (CO_2)	per cent.	6·34	9·45
Carbon monoxide (CO)	,,	—	—
Oxygen (O)	,,	13·00	14·09
Nitrogen (by difference) (N)	,,	80·66	76·46

22.	Average temperature leaving boiler flue	°F.	642
23.	Mean specific heat of products of combustion		0·249

Air and Draught.

24.	Temperature of outside air	°F.	62·7
25.	Barometric pressure (29·92 inches of mercury)	lb. per sq. in.	14·7
26.	Draught over fire	inches of water	0·2
28.	Draught at gas exit from boilers	,, ,,	0·22
29.	Draft at base of chimney	,, ,,	0·51

Feed-Water.

31.	From pump per hour	lb.	10,903
	Water evaporated per hour	lb.	10,901
32.	Temperature of feed to boilers	°F.	259

Steam.

33.	Gauge pressure	lb. per sq. in.	155·3
34.	Absolute pressure	,, ,,	170
36.	Temperature of saturation	°F.	368·2
	Temperature of steam leaving superheater	°F.	491·6

HEAT ACCOUNT AND DEDUCTIONS.

BOILER AND SUPERHEATER.

Heat Account (per lb. of dried fuel).

		B.Th.U.	Per cent.
37.	Total heat value of 1 lb. of dried fuel	14,040	—
37a.	Available heat (hydrogen in coal burned to steam)	13,649	100

Ref. No.		B.Th.U.	Per cent.
38.	Heat transferred to water and superheated steam (and thermal efficiency) . . .	8,173	59·89
39.	Heat carried away by products of combustion	1,658	12·15
40	Heat carried away by excess air . . .	2,628	19·26
41.	Heat lost in evaporating and in superheating moisture mixed with fuel	66·4	0·49
42.	Heat lost by incomplete combustion. . .	—	—
43.	Heat lost by unburned carbon in ash . .	482	3·53
44.	Balance of heat account. Errors of observation, and unmeasured losses, such as —those due to radiation, escape of unburned hydrocarbons, superheating moisture in air, loss in hot ashes, etc .	641·6	4·68
	Total of lines 38 to 44, equal to line 37a	13,649	100·00

Deductions.

45.	Heat transmited per sq. ft. of heating surface per hour	B.Th.U. 4992
46.	Weight of fuel fired per sq. ft. of grate per hour	lb. 19·158
47.	Weight of dried fuel fired per sq. ft. of grate per hour	lb. 18·27
48.	Water evaporated and superheated per lb. of fuel as fired	lb. 7·487
49.	Equivalent evaporation from and at 212 F. per lb. of fuel as fired	lb. 8·071
50.	Water evaporated and superheated per lb. of dried fuel	lb. 7·848
51.	Equivalent evaporation from and at 212 F. per lb. of dried fuel	lb. 8·46
52.	Equivalent evaporation from and at 212 °F. per lb. of carbon value of dried fuel .	lb. 9·02
53.	Weight of feed-water evaporated from and at 212 °F. per square foot of heating surface per hour	lb. 5·167
55.	Air used per lb. of dried fuel	lb. 29·6
56.	Ratio of air used to air theoretically required	2·81

Economizer.

57. General description of economizer :—Three groups of pipes 8 + 8 + 12 by 8 pipes broad. Total 224 pipes.

58. Heating surface of economizer, 2240 square feet.

Data Deduced from Observations.

Economizer.

61.	Weight of feed water per hour	lb. 10,903
62.	Temperature of feed into	°F. 60

Ref. No.

63. Temperature of feed out of . . . . °F. 258·8
64. Temperature of flue gases into . . . . . . °F. 642
65. Temperature of flue gases out of . . . . . °F. 326
66. Analysis of dry flue gases leaving economizer—

		By vol.	By weight.
Carbon dioxide (CO_2)	per cent.	5·39	8·06
Carbon monoxide (CO) . . .	,,	—	—
Oxygen (O)	,,	14·14	15·38
Nitrogen (by difference) (N) . .	,,	80·47	76·56

HEAT ACCOUNT (PER LB. OF DRIED FUEL).

Economizer.

	B.Th.U.	Per cent.
77. Heat received from boiler flues, in dry gases and steam, per lb. of dried fuel (reckoned from air temperature) . . .	4352	100
78. Heat transferred to the water (and efficiency of economizer)	1560	35·85
79. Heat carried off in chimney gases . . .	2331	53·56
80. Balance of heat account, including errors of observation and difference of heat contained in brickwork at beginning and end of test, etc.	461	10·59
	4352	100·00

Deductions.

85. Heat transmitted per sq. ft. of heating surface of economizer per hour . . . . B.Th.U. 968
87. Thermal efficiency of boilers, economizer and superheater combined . . . . per cent. 71·3

Exercises XI

1. A boiler produces 8·9 lb. of steam per lb. of coal from feed-water at 45° C. (113° F.). The pressure of the steam is 150 lb. per square inch absolute. What is the equivalent evaporation from and at 100° C. (212° F.) per lb. of coal, (*a*) on the assumption that the steam is dry and saturated, (*b*) on the assumption that the dryness fraction of the steam is 0·98?

2. The following particulars are taken from the reports of trials of three steam boilers A, B, and C:—

		Boiler.		
Absolute pressure of steam	lb. per sq. in.	50	155	200
Temperature of feed-water	°C.	15	60	20
	°F.	59	140	68
Dryness fraction of steam		1·00	0·98	—
Degrees of superheat in steam	C.	—		50
	F.	—		90
Steam produced per lb. of coal	lb.	8·7	9·5	9·3

Calculate for each boiler the equivalent evaporation from and at 100° C. (212° F.) per lb. of coal.

3. The following particulars relate to a trial of two marine boilers of the Scotch type.—Duration of trial, 16 hours. Total heating surface, 3160 square feet. Total grate area, 42 square feet. Mean absolute pressure of steam, 180 lb. per square inch. Total coal burned, 15,072 lb. Total weight of water evaporated, 137,856 lb. Temperature of feed-water, 106° F. Steam assumed to be dry and saturated.

Calculate, (1) the weight of coal burned per square foot of grate per hour, (2) the weight of water evaporated per lb. of coal under the actual conditions, (3) the equivalent evaporation from and at 212° F. per lb. of coal, (4) the equivalent evaporation from and at 212° F. per square foot of total heating surface per hour.

4. A locomotive boiler has a total heating surface of 1214 square feet, of which 132 square feet is fire-box surface. The heat transmitted to the water through the fire-box surface may be taken as 40 per cent. of the whole heat transmitted to the water. If the average evaporation for the whole heating surface is 10 lb. per square foot per hour, from and at 212° F., what is the evaporation per hour from the fire-box per square foot of fire-box heating surface, and what is the evaporation per hour from the remaining heating surface per square foot of that surface?

5. Using the rule of Art. 159, calculate the horse-power of a boiler which produces 18,942 lb. of steam in 6 hours from feed water at 239° F., the pressure of the steam being 155 lb. per square inch absolute. Assume that the steam is dry and saturated.

6. Calculate the efficiency of a steam boiler from the following data:—Coal burned per hour, 105 lb. Calorific value of coal, 7750 C.H.U. (13,950 B.Th U.). Feed water per hour, 806 lb. Temperature of feed water, 10° C. (50° F.). Absolute pressure of steam, 120 lb. per square inch. Dryness fraction of steam, 0·99.

7. A boiler produces 5000 lb. of saturated steam per hour at 160 lb. per square inch absolute pressure (total heat 1194·5 B.Th.U. per lb.), and the feed-water is heated by an economizer to a temperature of 250° F. (water heat 218·5 B.Th.U.). 550 lb. of coal of a calorific value of 13,500 B.Th.U. are fired per hour, and it is ascertained that 10 per cent. of the fuel is unburned. Find the thermal efficiency of the boiler, and also of the grate and boiler combined. [Inst. C.E.]

8. In a boiler test the following quantities were obtained:—Mean temperature of feed, 55° F. Mean boiler pressure, 165 lb. per square inch absolute. Mean steam dryness 0·95. Weight of coal burned per hour, 500 lb. Calorific value of coal, 14,000 B.Th.U. per lb. Weight of water supplied to boiler in 7 hours 14 minutes, 32,500 lb. The weight of water in the boiler at the end of the test was less than that at the commencement by 2000 lb.

Calculate the actual evaporation per lb. of coal, the equivalent evaporation from and at 212° F. per lb. of coal, and the thermal efficiency of the boiler. [U.L.]

9. In a boiler trial 3600 lb. of coal were consumed in 24 hours. The weight of water evaporated was 28,800 lb., and the mean steam pressure by gauge was 95 lb. per sq. in. The coal contained 3 per cent. of moisture and 3·9 per cent. of ash by analysis. The feed water temperature was 95° F. (35° C.). Calorific value of one pound of coal 13,000 B.Th.U. (7222 C.H.U.). Total heat of one pound of steam at 110 lb. per square inch absolute 1188 B.Th.U. (660 C.H.U.).

Determine the efficiency of the boiler, and the equivalent evaporation, (1) per lb. of dry coal, (2) per lb. of combustible. [U.L.]

10. During a boiler trial the following data were obtained:—

Temperature of steam	373° F.
Moisture in steam	1 per cent.
Temperature of feed water	55° F.
Water evaporated per lb. of coal	9·4 lb.
Calorific value of coal fired, per lb.	14,720 B.Th.U.
Rise in temperature of the flue gases	450° F.
Specific heat of products	0·24

Analysis of coal:—
C, 86 per cent.; H, 4 per cent.; O, 3 per cent.
Analysis of flue gases by volume:—
CO_2, 10·4 per cent.; O, 9·1 per cent.

Determine—
(a) The thermal efficiency of the boiler.
(b) The heat carried away by the flue gases per pound of coal fired. [U.L.]

11. The percentage composition of a dry coal is: C, 83; H, 5; O, 4; ash, 8. This coal is used in a boiler and 20 lb. of air are supplied per lb. of dry coal. The temperature of the air is 15° C. (59° F.), and the temperature of the chimney gases is 330° C. (626° F.). The higher calorific value of the dry coal is 8280 C.H.U. (14,904 B.Th.U.) per lb., and the lower calorific value is 8040 C.H.U. (14,472 B.Th.U.) per lb.

Find the heat carried away in the chimney gases per lb. of dry coal, expressing the result as a percentage of the calorific value of the coal, (a) allowing for the latent heat in the steam and taking the higher calorific value of the coal, (b) neglecting the latent heat in the steam and taking the lower calorific value of the coal.

Calculate also the mean specific heat of the mixture of chimney gases between 15° C. and 330° C., (1) on the assumptions (a), and (2) on the assumptions (b).

12 350 lb. of coal are burned on a boiler grate per hour, and 3000 lb. of water are passed through the economizer, the temperature of the water being raised 150° F. and the temperature of the waste gases being lowered 300° F.; if the specific heat of the flue gases is 0·24, what weight of furnace gases are produced per pound of coal burned? [Inst. C.E.]

13. A boiler fuel whose calorific value is 8120 C.H.U. (14,616 B.Th.U.) per lb. contains 85 per cent. of carbon. The chimney gases contain 11·5 per cent. of CO_2 and 0·5 per cent of CO by volume. 5 per cent. of the carbon of the fuel is lost in the ashes. Calculate the percentage loss of heat due to the formation of CO instead of CO_2.

14. In a boiler trial the analysis of the dry coal used was, per cent.: C, 84; H, 5; O, 8; ash, etc., 3. The moisture in the coal as used was 0·02 lb. per lb. of dry coal. The calorific value of the dry coal was 8190 C.H.U. (14,742 B.Th.U.).

The analysis of the dry chimney gases by volume was, per cent.: CO_2, 10·1; CO, 0·3; O, 9·3; and N (by difference), 80·3.

The temperature of the air was 25° C. (77° F.) and the temperature of the chimney gases was 315° C. (599° F.).

Calculate the following heat losses per lb. of dry coal stoked, expressing each loss as a percentage of the calorific value of 1 lb. of dry coal.
(a) Heat lost through incomplete combustion of carbon.
(b) Heat lost in steam in chimney gases, taking the mean specific heat of superheated steam as 0·48.
(c) Heat lost in dry products of combustion exclusive of excess air, taking the mean specific heat as 0·24.
(d) Heat lost in excess air, taking the mean specific heat as 0·24.

CHAPTER XII

RECIPROCATING STEAM ENGINES

171. The Reciprocating Engine Mechanism.—The principal parts, other than the valve gear, common to reciprocating steam engines are represented in Fig. 215, which illustrates a simple form of engine The parts to be mentioned may vary in form in engines of different sizes or by different makers. In subsequent chapters most of the details of reciprocating engines are separately considered and illustrated with a considerable degree of fullness.

Referring to Fig. 215, the *cylinder* 1 is bolted at one end to the *frame* 2. The *cylinder cover* 3 is bolted to the other end of the cylinder. The cylinder is covered with some form of non-conducting composition 4 called *lagging*, and a finished appearance is given to the cylinder by covering the lagging with planished steel sheet 5. The

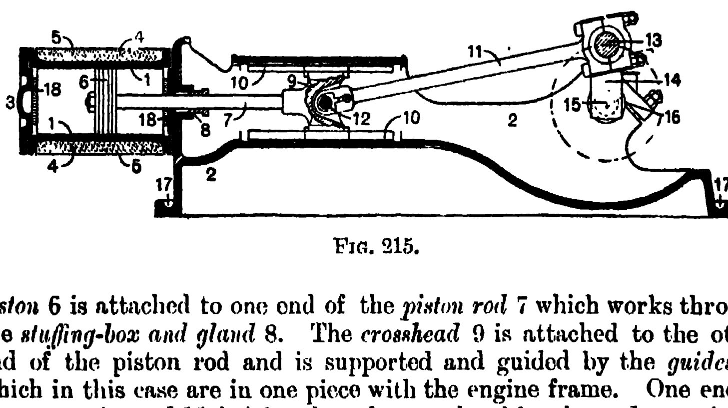


FIG. 215.

piston 6 is attached to one end of the *piston rod* 7 which works through the *stuffing-box and gland* 8. The *crosshead* 9 is attached to the other end of the piston rod and is supported and guided by the *guides* 10 which in this case are in one piece with the engine frame. One end of the *connecting rod* 11 is jointed to the crosshead by the *gudgeon* pin 12 while the other end embraces the *crank pin* 13 of the *crank* 14 which in this case is forged in one piece with the *crankshaft* 15. The crank shaft runs in the *main bearings* of which one is shown at 16. It is a good plan to form a *gutter* 17 round the base of the frame or bed of the engine to catch the waste lubricating oil and prevent it spreading over the floor of the engine room. Steam enters and leaves the cylinder by the *ports* 18.

Important details not shown in Fig. 215 are, the *valve* or *valves* and *valve gear* for the distribution of the steam in the cylinder, and the *governor* for regulating the speed, but these have separate chapters

devoted to them, as, also, have *air pumps* and *condensers* for condensing engines.

[illegible]. **Types of Reciprocating Engines.**—Of the numerous classifications which may be made of reciprocating engines the principal are the following:—

Single and Double Acting.—In a single-acting engine the steam acts on one side of the piston and the atmosphere on the other, while in a double-acting engine the steam acts on both sides of the piston. Other conditions being the same, a double-acting engine will develop twice the power of a single-acting engine, and for this reason single-acting steam engines are now rare.

Vertical and Horizontal.—An engine is said to be vertical or horizontal according as the axis of its cylinder is vertical or horizontal. A horizontal engine takes up more floor space than a vertical engine of the same size. The parts of a horizontal engine are generally more accessible than those of a vertical engine. The conditions in a vertical engine are more favourable to the piston and cylinder as regards wear and friction than they are in a horizontal one.

Slow and High Speed.—In speaking of the speed of a reciprocating engine it is the speed of the crank shaft, in revolutions per minute, which is referred to. An engine having a piston stroke of 5 feet and a shaft speed of 60 revolutions per minute would have an average piston speed of 600 feet per minute. This would be called a slow-speed engine. Another engine having a piston stroke of 1 foot and a shaft speed of 300 revolutions per minute would be called a high-speed engine but its average piston speed would be the same as that of the slow-speed engine just mentioned.

Speaking generally, an engine speed of less than 100 revolutions per minute would be called a slow speed, and a speed not less than 250 revolutions per minute would be called a high speed, while a speed between 100 and 250 revolutions per minute would be called a medium speed.

The great advantage of a high-speed engine over a slow-speed one is that for the same power it is much smaller, and several obvious advantages follow this reduction in dimensions. The high speed also conduces to economy of steam as there is less time for the interchange of heat between the steam and the cylinder.

Non-condensing and Condensing.—A non-condensing engine is one in which the steam exhausts into the atmosphere and the steam pressure in the cylinder should therefore not fall to less than that of the atmosphere. In a condensing engine the steam, after doing work in the cylinder, passes into a condenser and is reduced to water at a pressure considerably below that of the atmosphere. This enables the steam to do more work in the cylinder.

Simple and Compound.—A simple engine is one in which each of its cylinders receives steam direct from the boiler and exhausts into the atmosphere or into a condenser. In a compound engine the steam from the boiler after expanding to a certain extent in one cylinder, called the *high-pressure cylinder*, exhausts into a larger cylinder, called the *low-pressure cylinder*, where the expansion is completed. The low-pressure cylinder generally exhausts into a condenser, that is to say.

compound engines are generally condensing engines but they may be non-condensing.

This subdivision of the expansion may be carried out in three or even four cylinders in succession as in *triple-expansion* and *quadruple-expansion engines.* Whatever the number of expansion stages, the high-pressure cylinder is that in which the first stage is performed and the low-pressure cylinder is that in which the last stage is performed. The intermediate expansion takes place in the *intermediate cylinder* in the case of a triple-expansion engine and in a *first intermediate* and then in a *second intermediate cylinder* in the case of a quadruple-expansion engine.

In large engines, when a single low-pressure cylinder would be inconveniently large, two low-pressure cylinders are used, each taking half the steam discharged from the preceding cylinder.

Since the extension in the use of superheated steam and the introduction of the steam turbine, quadruple-expansion engines are not now so common as they were.

The reason for compounding is given in the next Art.

173. Reason for Compounding.—Unfortunately for the efficiency of reciprocating steam engines there is an interchange of heat between the steam and the cylinder. During the early part of the stroke when the steam is hottest it gives up heat to the cylinder. After cut off, as the steam expands, its temperature falls. The fresh surface of the cylinder exposed to the expanding steam, as the piston advances, has just been in contact with the steam on the other side of the piston, which is at the temperature corresponding to the back pressure, and, up to a certain point, at a lower temperature than the expanding steam. Consequently up to a certain point of the stroke the fresh surface of the cylinder will extract heat from the expanding steam. But the surface of the cylinder which the piston has left behind has already been heated up during the earlier part of the stroke and is hotter than the expanding steam and will give up heat to the steam. The net result is that up to some point in the stroke the steam is giving heat to the cylinder and beyond that point the cylinder is giving heat to the steam.

If the steam is saturated when it is giving heat to the cylinder some of the steam will condense, and when the cylinder is giving heat to the steam some of the water in it will be re-evaporated.

Stated briefly, the steam loses heat to the cylinder during the earlier part of the stroke and partly recovers this towards the end of the stroke, but the heat recovered is almost immediately and almost entirely lost in the exhaust. This interchange of heat will evidently be greater the greater the ratio of expansion with its greater range of temperature.

Now if the steam is transferred to another cylinder when only partly expanded, and the expansion be completed in that cylinder, the range of temperature in each cylinder will be much less than when all the expansion takes place in one cylinder. There will now be less loss of heat from the steam to the cylinder over the earlier part of the stroke and less gain of heat during the later part. Also, and this is most important, the heat recovered by the steam towards the end of

the stroke in the first cylinder is effective in doing work in the second cylinder.

In triple-expansion engines the principle of compounding is extended to three stages when the initial pressure and ratio of expansion are higher than would be suitable for compound engines.

The adoption of superheated steam has enabled economical engines to be constructed with higher initial pressures and greater expansion ratios with fewer expansion stages. Condensing engines are now made compound with initial pressures up to 160 lb. per square inch by gauge. The greater the superheat the higher may be the pressure without increasing the number of expansion stages. For initial pressures above 160 lb. per square inch, by gauge, triple-expansion engines are generally used.

In locomotives, which are non-condensing, superheating is now common and pressures up to 180 lb. per square inch, by gauge, are common in simple engines, and up to 225 lb. per square inch in compound engines.

Incidentally there are some further advantages due to compounding. When the different pistons are coupled to separate cranks this leads to a more uniform turning effort on the crank shaft. A more uniform turning effort could of course be obtained by using two or more simple cylinders of smaller size and two or more cranks, but the turning effort would not then be so uniform as in the equivalent compound or triple expansion engine with the same total ratio of expansion because the range of piston effort in the simple engine is much greater than in the cylinders of the others.

A reduction of the range of cylinder pressure also means a reduction in the stresses in the framing, shafting, etc.

174. Woolf and Receiver Compound Engines.—A compound engine may be a *Woolf engine* or a *receiver engine*. In the Woolf type the pistons of the H.P. and L.P. cylinders begin and end their strokes together; such is the case when the pistons are on the same rod and are coupled to one crank, or when the pistons are coupled separately to cranks at 180°. Other types are called receiver engines because a receiver or reservoir is necessary to receive the steam from the H.P. cylinder and hold it until it is wanted by the L.P. cylinder. In many cases the receiver is simply the pipe leading from the H.P. to the L.P. cylinder.

In a Woolf engine the time during which exhaust takes place from the H.P. cylinder coincides more or less with the time of admission to the L.P. cylinder. Hence no receiver, or only a small one, such as is formed by the pipe between the cylinders, is necessary.

The pressure in the receiver will of course fluctuate according to the supply of and demand for steam.

The receiver of a receiver engine is frequently jacketed, and the jacket supplied with steam direct from the boiler. The receiver may be multi-tubular, like a surface condenser. In that case the exhaust steam from the H.P. cylinder passes through the tubes and the space surrounding them is supplied with boiler steam.

The object in reheating the steam on its way from the H.P. to the

L.P. cylinder is to dry it so that it may be more effectively used in the latter cylinder.

175. Arrangement of Cylinders and Cranks in Compound and Triple-Expansion Engines.—A common form of compound engine is the *cross-compound* shown in Figs. 216 and 218. This is a receiver engine. There are two cranks at right angles.

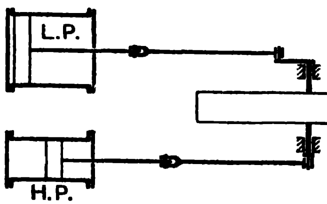


FIG. 216.—Cross-compound.

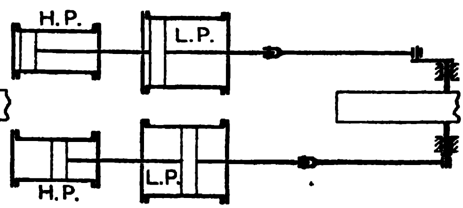


FIG. 217.—Twin tandem compound.

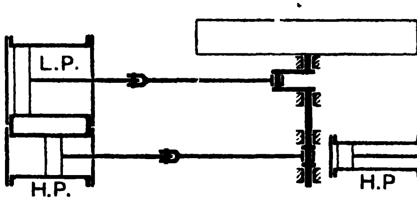


FIG. 218.—Cross-compound.

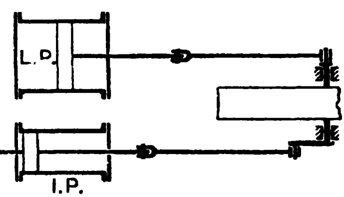


FIG. 219.—Triple-expansion.

A twin tandem compound engine is shown in Fig. 217. This engine is made up of two Woolf engines. The cranks are at right angles. A three-cylinder triple-expansion engine is shown in Fig. 219. There are two cranks at right angles, the H.P. and I.P. pistons being coupled to one and the L.P. piston to the other.

The cylinders for a three-cylinder three-crank triple-expansion engine are shown in Fig. 220. The cranks make 120° with one another.

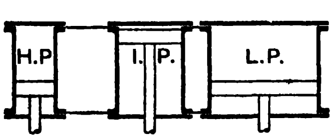


FIG. 220.—Three-cylinder triple-expansion.

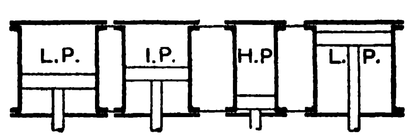


FIG. 221.—Four-cylinder triple-expansion.

The cylinders for a four-cylinder four-crank triple-expansion engine are shown in Fig. 221. This is an arrangement which lends itself to proper balancing by adjusting the angles between the cranks and the magnitudes of the reciprocating masses. The cranks of the right hand pair of cylinders are more or less opposite to one another and so are those of the left hand pair, and the right hand pair of cranks are approximately at right angles to the left hand pair.

Another arrangement of the cylinders of a four-cylinder triple-expansion engine is shown in Fig. 222. In this case there are two cranks at right angles.

176. The Allen High-Speed Triple-Expansion Engine.—As an example of a modern high-speed engine which has proved to be highly efficient, reliable and durable, the type manufactured by Messrs. W. H. Allen, Son & Co., Bedford, may be taken. The engine as a whole, which is of the three-cylinder triple-expansion type, is represented in Fig. 223. Illustrations and descriptions of various details of this engine will be found further on and an index to these is given at the end of this Art.

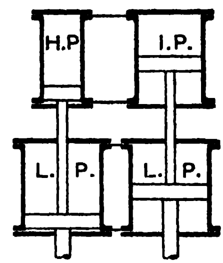


FIG. 222.—Four-cylinder triple-expansion.

In the particular engine shown in Fig. 223 the diameters of the cylinders are: high pressure (HP), 14 inches; intermediate pressure (IP), 21 inches; low pressure (LP), 32 inches. All the pistons have the same stroke, 13 inches. The valves which control the admission and exhaust of the steam to and from the cylinders are of the piston tpye, made of cast iron, and without packing rings.

Steam from the boiler, after passing through the throttle valve, enters the high-pressure valve chest at A and after doing its work in the high-pressure cylinder exhausts over the ends of the valve to the outlet B from which it is led through a passage at the back of the cylinder to the inlet C of the intermediate-pressure valve chest. After doing its work in the intermediate-pressure cylinder the steam exhausts through D and the passage DE behind the cylinder to the low-pressure valve chest. After doing its work in the low-pressure cylinder the steam is led through F to the condenser which is separate from the engine.

An important feature of this and other high-speed engines is the system of forced lubrication. The lubricant, which should be a pure mineral oil, is supplied to the various bearing surfaces by a small pump at a pressure of about 15 lb. per square inch. The oil pump is of simple design, having neither loose valves nor packing, and is driven off the high-pressure eccentric. The oil, which is pumped from the bottom of the crank chamber, is first delivered through steel pipes to an oil strainer, and thence to a main distributor which acts as a reservoir. An air vessel is fitted to this distributor in order to keep the oil pressure nearly constant. From the distributor the oil is forced through pipes direct to each of the main bearings

Holes drilled in the crank shaft, shown by dotted lines in Fig. 223, deliver a supply of oil to the crank pins and eccentric sheaves from the main bearings. From each crank pin a pipe running up the connecting rod leads the oil to the crosshead pin and thence through a hole to the guide shoe. The valve rod pins and valve rods are lubricated in a similar manner through pipes from the eccentrics.

The oil after passing through the various bearings is returned to the bottom of the crank chamber to be used over again. The piston rods and valve rods pass through glands 14 and 15 which scrape the

HP. High-pressure cylinder.
IP. Intermediate-pressure cylinder.
LP. Low-pressure cylinder.
1. Pistons.
2 Piston rods.
3. Crossheads.
4. Crosshead guides.
5. Connecting rods.
6. Crank shaft.
7. Main bearings.
8. Eccentrics.
9. Eccentric rods.
10. Piston valve rods.
11. Piston valves.
12. Piston rod stuffing-boxes.
13. Valve rod stuffing-boxes.
14. Piston rod oil-scraping glands.
15. Valve rod oil glands and guides.
16 Fly-wheel.
17. Governor.
18. Throttle valve.
19. Rod connecting governor with throttle valve.
20. Oil pipes from oil pump to main bearings.
21. Oil pipes from crank pins to crosshead pins.
22. Oil pipes from eccentrics to valve rod pins.
23. Balance weights.

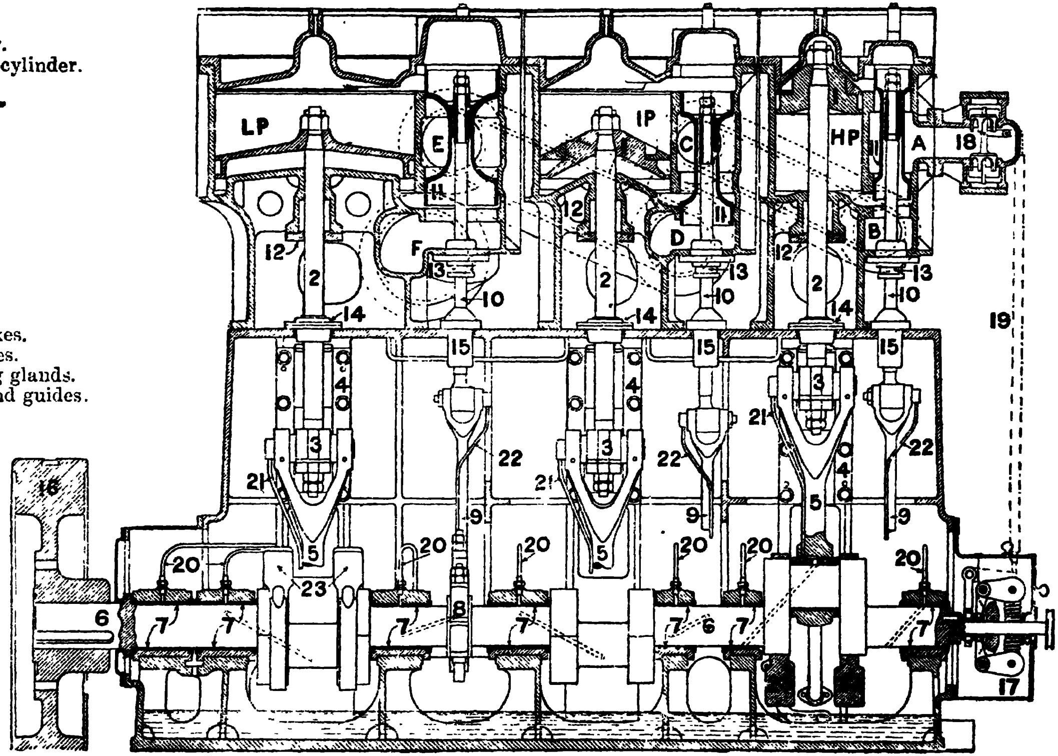


FIG. 223.—Allen high-speed enclosed triple-expansion engine.

oil off these rods and prevent it passing up to the cylinders. The internal lubrication of the cylinders is effected by means of a sight feed lubricator which delivers into the steam a regular and measured supply of oil.

With a boiler pressure of 150 lb. per square inch, a vacuum of 26 inches of mercury, and at a speed of 330 revolutions per minute, this engine, at full load, develops 670 indicated horse-power and 625 brake horse-power, with a steam consumption of $14\frac{1}{4}$ lb. per indicated horse-power per hour.

For full particulars of some of the details of this engine see Fig. 268, p. 256 (piston rod stuffing-box); Fig. 267, p. 256 (valve rod stuffing-box); Fig. 269, p. 256 (piston rod oil-scraping gland); Fig. 274, p. 258 (crosshead); Fig. 280, p. 261 (connecting rod); Fig. 300, p. 267 (eccentric, eccentric rod, etc.); Fig. 303, p. 269 (throttle valve); Fig. 406, p. 334 (governor); and Fig. 434, p. 347 (piston valve).

177. Reciprocating Marine Engines.—Slow-speed cargo boats, which form the most numerous class of ships afloat, require large

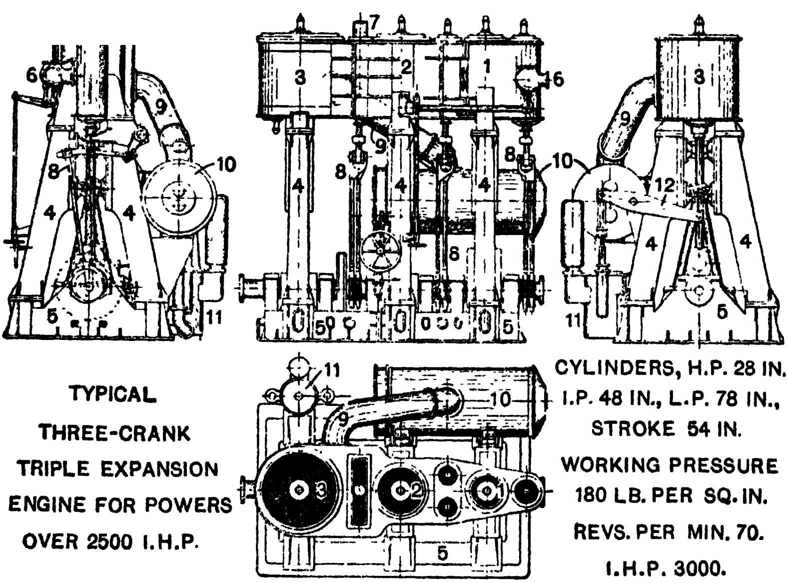


Fig. 224.

propellers rotating at speeds of 60 to 80 revolutions per minute. For this service reciprocating engines are almost exclusively used. The steam turbine has practically displaced the reciprocating engine in high-speed vessels and with the successful application of reducing gears will probably become more common in lower speed ships.

Of reciprocating engines on board ship the most common is the triple-expansion type. The general features of a three-cylinder three-crank triple-expansion marine engine are shown in Fig. 224. This is an illustration prepared from one given in a paper by Mr. D. Gibson

on "The Reciprocating Engine in Marine Practice and its Probable Future" in the *Transactions of the Manchester Association of Engineers*, 1908–9. The type illustrated is usually made for powers of 2500 to 4000 I.H.P.

Referring to Fig. 224, 1, 2, and 3 are the H.P., I.P., and L.P. cylinders respectively. The cylinders are bolted together and to the cast-iron columns 4 which are in turn bolted to the bed plate 5 which contains the bearings of the crank shaft. Steam enters the H.P. valve chest through the stop valve 6. The H.P. cylinder has a piston valve and the I.P. cylinder has two piston valves which move together, their spindles being connected to a crosshead beneath the valve chest. The L.P. cylinder has a double-ported slide valve fitted with a relief

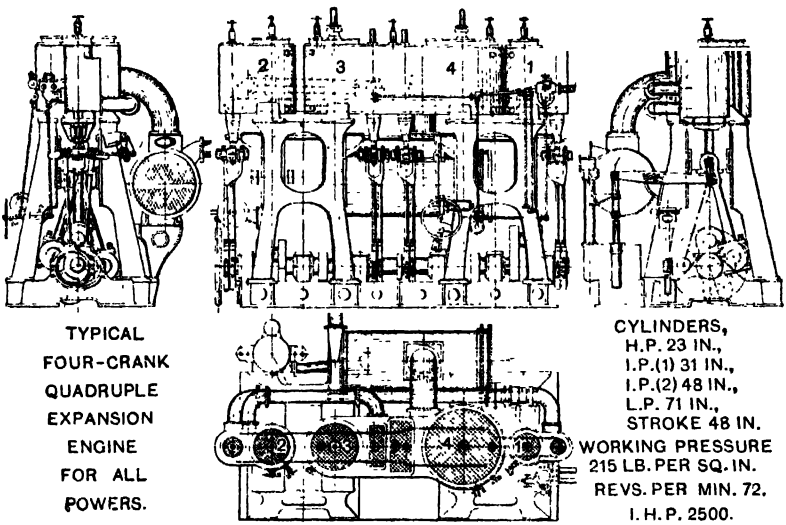


FIG. 225.

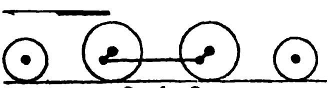
frame and having a balance cylinder 7 similar to that shown in Fig. 430, p. 345.

Each of the three valve gears 8 is a Stephenson link motion. The exhaust pipe 9 leads the steam from the L.P. cylinder to the surface condenser 10. The air pump 11 is driven by rocking levers 12 from the L.P. crosshead. It is now usual to have separate and independent circulating and feed pumps, the circulating pump being of the centrifugal type.

The four-cylinder four-crank triple-expansion engine has its cylinders arranged as in Fig. 221, p. 235. This type permits of a higher speed, if necessary, on account of the more perfect balance of the reciprocating parts.

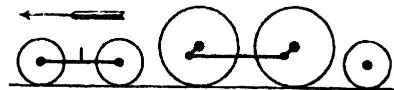
A typical four-cylinder four-crank quadruple-expansion engine is shown in Fig. 225; this illustration has also been prepared from one in Mr. Gibson's paper. 1, 2, 3, and 4 are the H.P., first I.P., second I.P., and L.P. cylinders respectively.

178. Wheel Arrangements of Locomotives.—The wheels of a locomotive are either *coupled wheels* or simply *carrying wheels;* of the former one pair, on the driving axle, are *driving wheels.* All the coupled wheels must be of the same diameter; the carrying wheels are always smaller than the coupled wheels but any two wheels on the same axle must be of the same diameter. Wheels in front are called *leading wheels* and those in the rear *trailing wheels.* Carrying wheels



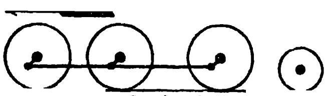
2–4–2

FIG. 226.



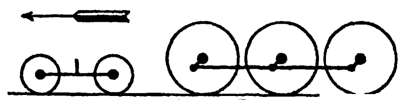
4–4–2

FIG. 227.



0–6–2

FIG. 228.

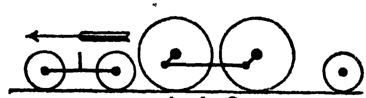


4–6–0

FIG. 229.

4–4–0

FIG. 230.

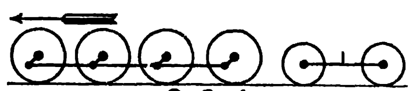


4–4–2

FIG. 231.

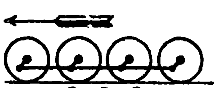
0–6–0

FIG. 232.



0–8–4

FIG. 233.



0–8–0

FIG 234.

4–6–2

FIG. 235.

2–8–8–2 (ARTICULATED)

FIG. 236.

may be in separate pairs or four of them may be on two axles whose bearings are in a frame forming a *four-wheeled bogie.*

The *wheel-base* of a locomotive is the distance between the axes of the front and rear wheels. The *rigid wheel* base is the distance between the axes of the forward and rear coupled wheels.

Flexibility in running round curves is obtained in various ways. A four-wheeled bogie is capable of turning slightly about a vertical pin placed at or near the centre of the bogie. A two-wheeled bogie

is also capable of turning about a vertical pin at some distance from the axle. A *radial axle-box* enables a single axle to have the same kind of movement as that of a two-wheeled bogie. Arrangements may also be provided on bogies and single axles of carrying wheels for allowing side play.

To those familiar with locomotives a good deal of information is conveyed by a statement of the wheel arrangement. This statement should give the number of coupled wheels, the number of wheels in front of the coupled wheels and the number behind. By means of a simple notation this may be done very briefly. Let a be the number of wheels in front of the coupled wheels, b the number of coupled wheels, and c the number of wheels behind, then a–b–c is a statement of the wheel arrangement. Examples are given in Figs. 226 to 236.

An *articulated* locomotive is one in which there are separate engine frames each with its own cylinders, valve gears, wheels, etc., each frame as a whole being capable of slight angular movement with respect to the boiler in order to give flexibility in rounding curves. Articulated locomotives are used when great power combined with exceptionally large tractive force is required.

179. Tractive Force of a Locomotive.—Neglecting the engine friction the work done in the cylinders in a given time is equal to the work done at the drawbar in the same time. Taking the case of the ordinary two-cylinder engine, let d = diameter of cylinders in inches, l = stroke of pistons in inches, p = mean effective pressure in cylinders in lb. per square inch, D = diameter of driving wheels in inches, and T = tractive force or drawbar pull in lb.

In one revolution of the driving wheels the work done in the cylinders is $\frac{\pi}{4}d^2pl \times 2 \times 2$ inch-pounds.

The distance travelled by the engine for one revolution of the driving wheels is πD inches, and the work done at the drawbar is πDT inch-pounds.

Hence, $\pi \mathrm{DT} = \frac{\pi}{4}d^2pl \times 2 \times 2$. Therefore $\mathrm{T} = \frac{d^2pl}{\mathrm{D}}$.

For starting the engine, p is generally taken at from 75 to 85 per cent. of the boiler pressure and for working at full power p may be taken at from 60 to 65 per cent. of the boiler pressure, the boiler pressure being the gauge pressure, not absolute pressure.

Allowing for the friction of the engine the actual tractive force is about 80 per cent. of the theoretical.

180. Adhesion of Locomotives.—When the driving wheels are just on the point of slipping on the rails the ratio of the tractive force at that instant to the total weight on the driving wheels (including the coupled wheels) gives the coefficient of friction or coefficient of adhesion between the wheels and the rails. This coefficient varies from about one-fourth (560 lb. per ton) when the rails are very dry or very wet to about one-ninth (250 lb. per ton) when the rails are damp and greasy.

It is usual to assume that when the engine is running the coefficient of adhesion is about one-sixth (370 lb. per ton) but at starting it may be taken at one-fourth (560 lb. per ton).

The adhesive force is the total weight on the rigid wheel base multiplied by the coefficient of adhesion.

In order that there may be no slipping of the wheels on the rails the adhesive force must exceed the actual tractive force.

181. **Express Locomotive: G.E.R.**—For express passenger traffic the four-coupled type of engine was at one time the most common but for the increasing loads which this traffic has demanded the six-coupled has become a very common type. As an example of a modern express passenger locomotive the following particulars of the 4–6–0 type used on the Great Eastern Railway may be of interest.

Boiler. The barrel has a mean inside diameter of 5 ft., and the length between the tube plates is 12 ft. 10 in. There are 191 steel tubes $1\frac{3}{4}$ inches external diameter, and 21 steel tubes $5\frac{1}{4}$ inches external diameter containing the elements of a Schmidt superheater. The tubes of the superheater elements have an external diameter of $1\frac{3}{8}$ inches.

The fire-box is of the Belpaire type. The outside measurements of the fire-box casing are: Length, 8 ft. 6 in.; width at top, 5 ft. $3\frac{3}{8}$ in.; width at bottom, 4 ft. $0\frac{1}{2}$ in. The front half of the grate slopes upwards to clear the axle of the intermediate coupled wheels while the back half is horizontal.

The total heating surface is 1919 sq. ft. made up as follows: Ordinary $1\frac{3}{4}$ in. tubes, 1123 sq. ft.; $5\frac{1}{4}$ in. tubes, 366·1 sq. ft.; superheater element tubes, 286·4 sq. ft.; fire-box, 143·5 sq. ft. The grate area is 26·5 sq. ft.

There are four 3-in. safety valves, and the working pressure is 180 lb. per sq. in.

Engine. There are two cylinders 20 inches in diameter. Stroke of pistons, 28 in. The steam distribution is by piston valves, one to each cylinder, each being driven through a Stephenson link motion by two eccentrics, one for forward and the other for backward running.

Fig. 237 shows, in plan and elevation and various sections, the principal parts of one half of the engine. The other half is the same as that shown but the crank of the one is at right angles to the crank of the other.

The principal parts are numbered and are here tabulated.

References are also given to illustrations of certain details which in some cases, although similar, may not be exactly the same as those on this particular engine.

1. Cylinder.
2. Piston.
3. Piston rod.
4. Piston tail-rod.
5. Piston rod stuffing-box with metallic packing (Fig. 264, p. 255).
6. Piston tail-rod guide (Fig. 265, p. 255).
7. Crosshead (Fig. 279, p. 260).
8. Slide bar.
9. Connecting rod (Fig. 284, p. 263).
10. Crank arms.
11. Crank axle.
12. Forward eccentric.
13. Backward eccentric.
14. Forward eccentric rod.
15. Backward eccentric rod.
16. The link, jointed to outer ends of eccentric rods.

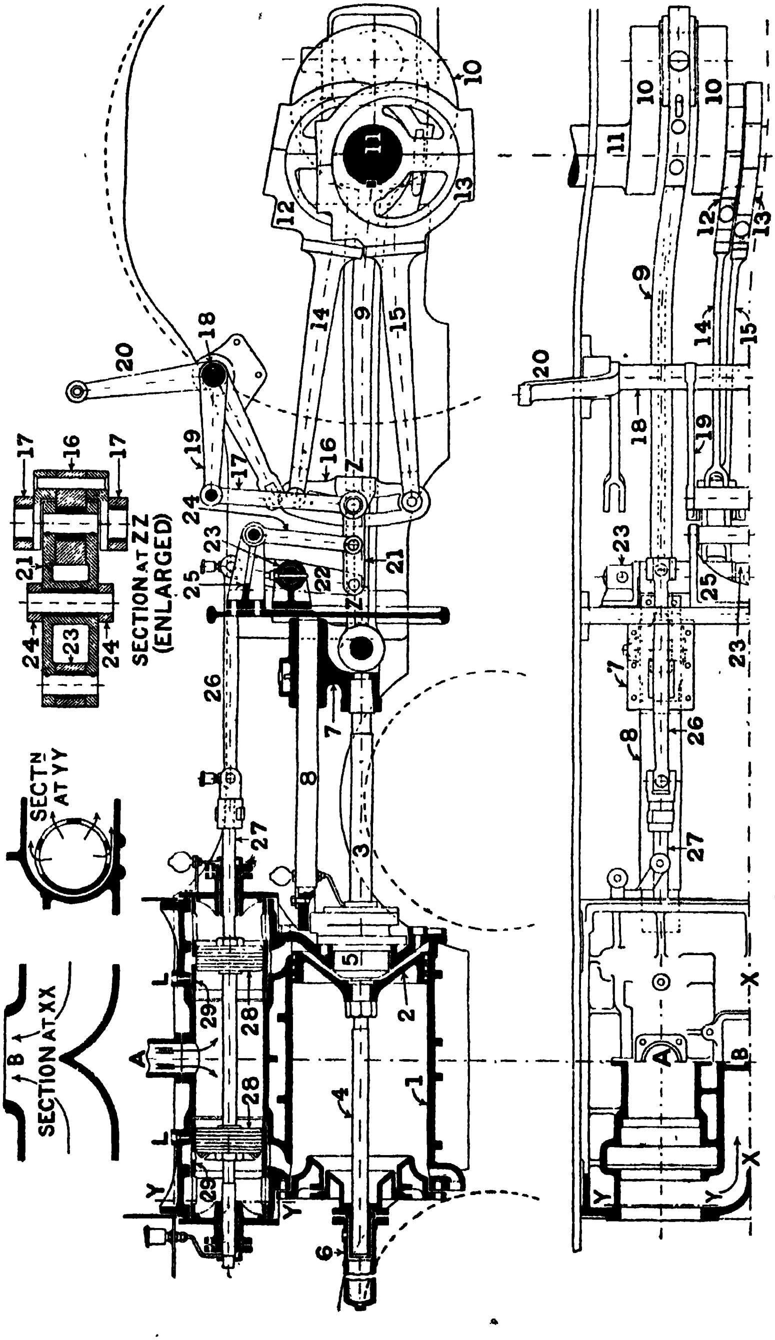


FIG 237.—4-6-0 Express passenger engine, G.E.R.

17. Suspension links supporting the link 16.
18. Weigh-bar shaft.
19. Arm forged on 18 and jointed to 17.
20. Arm forged on 18 for operating weigh-bar shaft through a rod jointed to its outer end and proceeding to the cab.
21. Link carrying motion block which works inside the link 16 and connects it to 22.
22. Lever rocking on the fixed axle carried by 23.
23. Brackets.
24. Suspension links for 21.
25. Bracket carrying 24.
26. Rod connecting lever 22 with 27.
27. Piston valve spindle.
28. Pistons of piston valve (Fig. 435, p. 347).
29. Liners in which piston valve works (Fig. 436, p. 348).

The steam from the boiler after passing through the superheater enters the valve casing at A between the two pistons of the piston valve and after passing through the ports in the liner 29 enters the cylinder. The steam, after doing its work in the cylinder, returns through the steam ports in the liner 29 and then passes through the outer ports in the liner to the blast pipe which is mounted at B. The course of the exhaust steam is clearly shown in the section on the plan, in which the liner 29 is removed, and in the sections at YY and XX.

Wheels and Wheel Base. The driving and coupled wheels have a diameter of 6 ft. 6 in. The wheels of the four-wheeled bogie have a diameter of 3 ft. 3 in. The tender has six wheels 4 ft. 1 in. in diameter

The total wheel base of the engine is 28 ft. 6 in. The rigid wheel base is 14 ft. and the bogie wheel base is 6 ft. 6 in.

The tender wheel base is 12 ft. and the total wheel base of the engine and tender is 48 ft. 3 in.

Weights in Working Order. The total weight of the engine and tender is 103·25 tons distributed as follows. On bogie wheels, 20 tons; on driving wheels, 16 tons; on intermediate wheels, 14 tons; on trailing wheels, 14 tons; on tender wheels 39·25 tons.

The capacity of the tender for coal is 4 tons and for water 3700 gallons.

The adhesive power of the engine at 508 lb. per ton is

$$508\,(16 + 14 + 14) = 22{,}352 \text{ lb.}$$

The tractive force, taking the mean effective pressure in the cylinders as 82 per cent. of the boiler pressure, is by the formula proved in Art. 179.

$$T = \frac{d^2pl}{D} = \frac{20^2 \times 0{\cdot}82 \times 180 \times 28}{12 \times 6{\cdot}5} = 21{,}194 \text{ lb.}$$

182. Semi-Portable Steam Power Plants.—The term *semi-portable* is applied to steam power plants in which the engine and boiler form one structure with one foundation. Such plants occupy comparatively small floor space and require only light and inexpensive foundations. There are two principal types of semi-portable steam plants.

(1) The *under-type* in which the boiler is of the locomotive type standing on a bed-plate or frame to which the engine is attached

under the barrel of the boiler. The engine is horizontal and usually non-condensing.

(2) The *over-type*, generally known as the *locomobile*, in which the engine is fitted on top of the boiler which is of the Cornish multitubular type. The locomobile has been developed to a much greater extent than the under-type in the direction of high thermal efficiency. The locomobile is generally condensing and has a superheater and a feed-water heater.

Fig. 238 is a general view of the locomobile as made by Messrs. Marshall, Sons & Co., Ltd., Gainsborough. The boiler 1, it will be seen, is cylindrical and horizontal; it has one furnace from which the

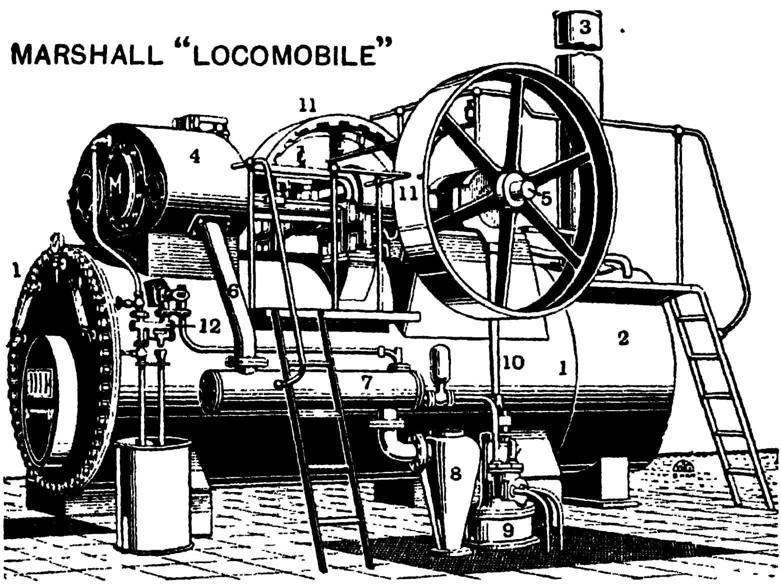


FIG. 238.

hot gases pass through a considerable number of horizontal tubes to the smoke-box 2 which is an extension of the boiler shell. The gases then pass up the chimney 3. There is a superheater in the smoke-box through which the steam passes on its way to the engine. The front end plate of the boiler is bolted to the shell and the back tube plate is also secured by bolts so that the furnace and tubes may readily be removed bodily for cleaning or repair.

The engine is of the two-cylinder compound type. The two cylinders 4 are side-by-side in one casting, fitted with liners, and are bolted directly to the boiler. Both cylinders have piston valves driven through rocking levers by eccentrics on the crank shaft 5. The crank shaft is supported in bearings carried by a steel saddle riveted to the boiler shell, the bearings being connected to the cylinder casting by a frame. The forces transmitted from the pistons to the crank shaft are

balanced in the frame and not in the boiler shell. Provision for expansion of the boiler in relation to the engine is also made.

The engine is controlled by a governor on the crank shaft which regulates the cut-off in the high-pressure cylinder.

The exhaust steam from the low-pressure cylinder is led by the pipe 6 to a tubular feed-water heater 7 and then passes to the jet condenser 8. The air pump 9 is worked by an eccentric on the crank shaft through the rod 10 which also connects to the feed pump. An injector 12 is also provided.

The power is taken off by belt from one or both of the fly-wheels 11.

Engines of the locomobile type are usually not of large power, say, 50 to 300 horse-power, but they have been made up to 1000 horse-power.

The close proximity of the engine to the boiler conduces to the reduction of radiation losses and the plant as a whole is highly efficient. The following are some particulars of a four hours' trial of a Marshall locomobile reported in *Engineering*, Nov. 6, 1914 :—Temperature of feed, 120° F. Steam pressure (gauge), 188·7 lb. per square inch. Superheat, 158° F. Vacuum, 27 inches of mercury. Revolutions per minute, 206. Brake horse-power, 124. Coal consumption, 1·47 lb. per B.H.P. per hour. Calorific value of coal, 11,500 B.Th.U. per lb. Coal burnt per square foot of grate, 23·7 lb. per hour. With good Welsh coal having a calorific value of 14,250 B.Th.U. per lb. the equivalent consumption would have been 1·185 lb. per B.H.P. per hour.

183. The Uniflow Engine.—What is now known as the *uniflow* or *straight-flow* steam engine was anticipated in a design of engine for which a British patent was granted in 1886 to Mr. L. J. Todd of London, and which he called the *mid-cylinder exhaust engine.* There

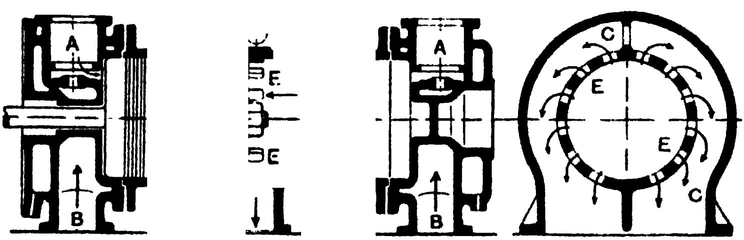


Fig. 239.

seems to be no record of Mr. Todd's design having been carried into practice at the time, but about twenty years later Professor Stumph of Charlottenburg resuscitated the mid-cylinder exhaust engine and proved that, if properly designed, it could be a most efficient engine. Since then its manufacture has been undertaken in many countries. The reason for calling this type a uniflow or straight-flow engine will appear presently.

The special features of the uniflow engine will be understood by reference to Fig. 239 which shows a longitudinal section and a central

transverse section of the cylinder. The ends or heads of the cylinder are hollow and each contains a double beat drop valve at A, but the valves are not shown. Steam enters the cylinder heads at B. At the middle of its length the cylinder barrel has a number of exhaust ports E which lead into a hollow belt C surrounding the barrel. The piston P is very long, its length being equal to its stroke minus the length of the exhaust ports. Steam entering the cylinder through the admission valve is cut off at an early part of the stroke and then expands as the piston completes its forward stroke. Exhaust takes place during the time that the exhaust ports E are uncovered by the piston. On the completion of the exhaust the steam remaining in the cylinder is highly compressed during the return stroke of the piston.

In an ordinary reciprocating engine the steam after following the piston during one stroke has to return and leave the cylinder by the port through which it entered at the end of the cylinder or by another port at the same end. In the mid-cylinder exhaust engine there is no such return of the steam, hence the designation uniflow or straight flow applied to this type of engine.

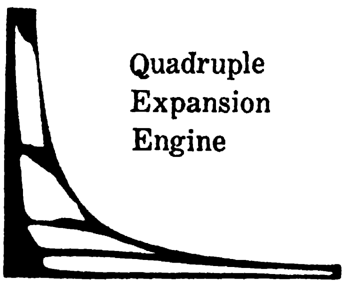


FIG. 240.

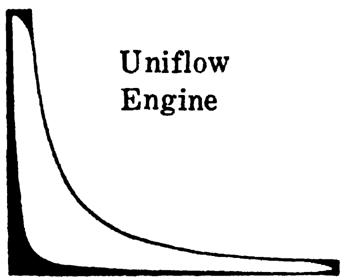


FIG. 241.

As has already been explained in giving the reason for compounding, the whole of the cylinder of an ordinary reciprocating engine is swept by the exhaust steam, and the fresh steam on entering loses a large amount of heat to the cooled cylinder: but in the uniflow engine the cylinder in the neighbourhood of the exhaust ports remains at the exhaust temperature while the ends of the cylinder remain practically at the initial steam temperature. During expansion the steam will of course fall in temperature and some of it will condense, but at the admission end the expanding steam will receive heat from the cylinder head which is always full of steam at the initial temperature. The coolest steam will therefore be nearest to the piston and it is this steam which will be first exhausted, and when the exhaust ports are closed the steam shut into the cylinder will be practically dry and during its compression it will become superheated and the fresh supply of steam coming in at the beginning of the next stroke will enter the hot clearance space and mix with the hot compressed steam and there is therefore no initial condensation. The result is that a high ratio of expansion may be adopted in the cylinder of a uniflow engine without serious loss due to condensation and, consequently, this engine, with

all the expansion taking place in one cylinder, may be as economical as a multi-stage expansion engine in which the steam is expanded in several cylinders in succession.

The claim made for the uniflow engine in regard to economy of steam compared with the multi-stage expansion engine is exhibited graphically by the indicator diagrams in Figs. 240 and 241 given by Professor Stumph. Fig. 240 shows the diagrams from the cylinders of a quadruple-expansion engine combined in one diagram while Fig. 241 shows the diagram from a uniflow engine. The total ratio of expansion is the same in both cases. The black areas represent the losses.

To sum up the claims for the uniflow engine: The same amount of steam will do as much work in one cylinder of a uniflow engine as would be done in the several cylinders of a multi-stage expansion engine, the stroke volume of the one cylinder of the former being no larger than that of the low-pressure cylinder of the latter.

The design of drop valve used in Professor Stumph's engines is shown in Fig. 242. A roller in a slot in the reciprocating piece S acts on the cam shown which is connected to the valve spindle.

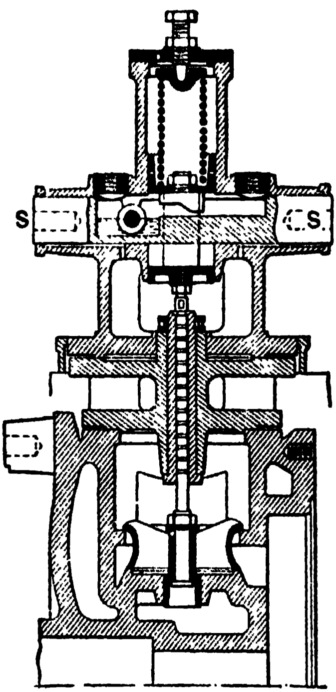


FIG. 242.

It must be noted that the cylinder of a uniflow engine has to be thicker than any of the cylinders of a multi-stage expansion engine of the same power, at the same speed, to withstand the high initial pressure, and the reciprocating parts are very heavy, causing large inertia forces at each end of the stroke but these forces are largely balanced by the high compression and high initial pressures.

CHAPTER XIII

RECIPROCATING STEAM ENGINE DETAILS

184. Pistons.—The points to consider in regard to pistons are:—(1) The form and material of the body of the piston, (2) the attachment of the body to the piston rod, and (3) the design of the packing which prevents or reduces leakage past the piston and yet enables the piston to move in the cylinder with comparatively little friction.

Hollow or box-shaped pistons are shown in Figs. 243, 244, and 247. These are made of cast iron, and the larger sizes are strengthened and stiffened by internal radial ribs. For certain forms of packing a *junk ring* J, Fig. 247, is necessary. The packing is placed in the annular space K between the junk ring and the flange F on the piston. The junk ring may be bolted to the piston as shown in Fig. 247, but more generally as shown in Fig. 249. Solid cast iron bodies are also used for pistons of small and medium diameter. Two forms of these are shown in Figs. 245 and 246.

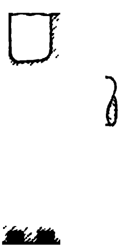


FIG. 243.

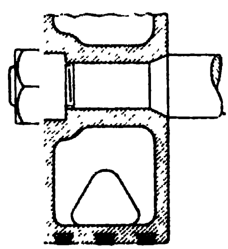


FIG. 244.

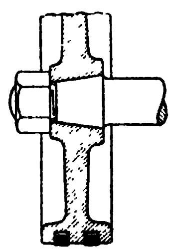


FIG. 245.

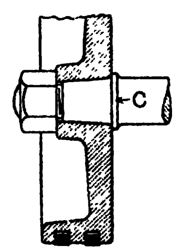


FIG. 246.

As a material for piston bodies cast iron is now very largely superseded by cast steel except for pistons of small diameter. Forged steel pistons are also used to a considerable extent. For the same strength and stiffness cast steel or forged steel pistons are much lighter than those made of cast iron. The box shape is never used for cast steel or forged steel pistons unless they are built up. Fig. 248 shows a cast steel piston from locomotive practice. A type of cast steel piston largely used, especially in marine engines, is shown in Fig. 249. The junk ring J is however generally made of cast iron.

The illustrations already referred to also show approved methods of connecting the piston to the piston rod. The part of the rod within the piston is generally entirely conical or partly conical and partly

cylindrical. A collar C on the piston rod, Figs. 246 and 249, at the base of the conical part, gives a shoulder which limits the bursting action of the conical part on the piston boss.

Referring to Fig. 249, the nuts on the junk ring studs S are locked by the ring L which is fastened to the junk ring by studs T. The collars on the studs S under the junk ring prevent these studs from

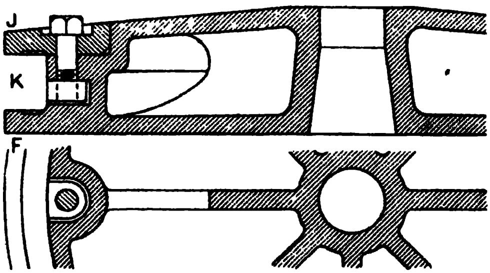


FIG. 247.

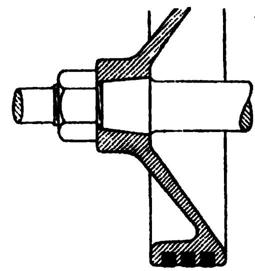


FIG. 248.

unscrewing. The studs for the locking ring are prevented from unscrewing by having square necks, inside square holes in the ring, and the nuts on the ring are secured by split pins through the tops of the studs The nut on the piston rod is secured by the locking plate R and two or three studs and nuts N.

It will be seen that the boss of the piston in Fig. 249 has a screw thread V cut on it. This is for the purpose of carrying an appliance,

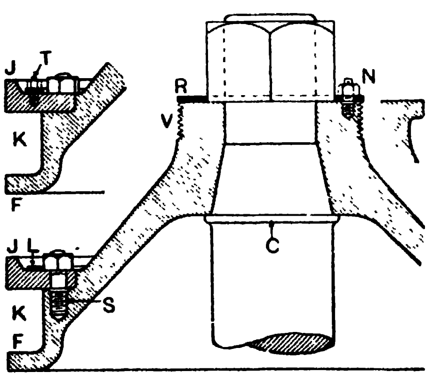


FIG. 249.

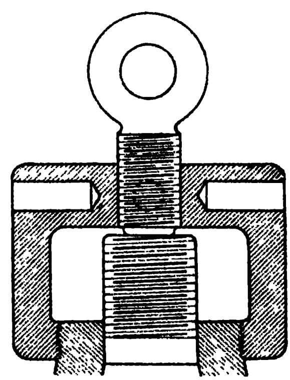


FIG. 250.

such as is shown in Fig. 250, for removing the piston from the rod when necessary. Instead of a screw thread V a narrow flange E is frequently formed on the boss for the same purpose. It may be left as an exercise for the student to design an appliance for removing the piston from the rod when the boss is flanged instead of being screwed.

The form of piston packing which is used more than any other is that first introduced by the late Mr. John Ramsbottom in the locomotives on the London and North Western Railway. In its simplest

form this packing consists of two or more rings, generally of cast iron, of rectangular section which fit into grooves turned in the rim of the piston. These rings are first turned a little larger in diameter than the bore of the cylinder (about $\frac{1}{4}$ inch per foot of diameter). They are then cut across at one point, either square or obliquely, as shown in Fig. 251, and sprung over the piston into the grooves prepared for them. Their own elasticity causes the rings to press outwards against the cylinder. The rings exert a more uniform pressure on the cylinder when, after they are cut, they are compressed and clamped between discs and turned to the exact diameter of the cylinder.

At the point where the ring is split a leakage of steam will take

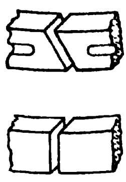


FIG. 251.

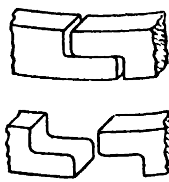


FIG. 252.

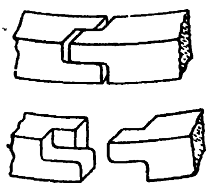


FIG. 253.

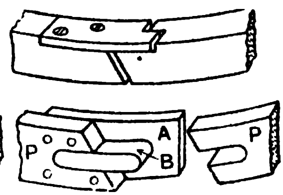


FIG. 254.

place but with quick running pistons this leakage is not of great importance. By cutting the rings so that the ends may overlap as shown in Fig. 252 this leakage is diminished. Cutting and overlapping the ends as shown in Fig. 253 makes a steam-tight joint if it is accurately constructed. Leakage at the split may be stopped by attaching an overlapping piece fitting into a recess on the side of the ring as shown in the upper part of Fig. 254. The lower part of Fig. 254 shows how leakage at the split of a broad packing ring may be prevented. The packing ring P has a slot cut in it where it is afterwards to be cut across. A gun-metal piece A having a tongue B cast

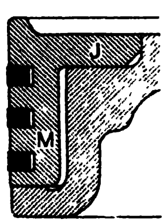


FIG. 255.

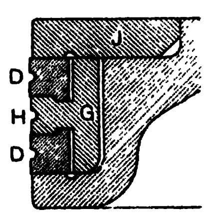


FIG. 256.

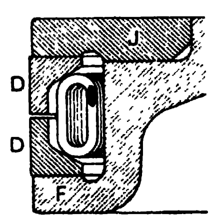


FIG. 257.

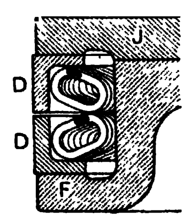


FIG. 258.

on it is carefully fitted on to the inside of the packing ring, the tongue B being fitted into the slot in the ring. After the ring is cut the piece A is fixed to it by several screws as shown.

Fig. 255 shows a cast iron carrier ring M in which grooves for the packing rings are turned. As shown in Fig. 255 this ring is cast with the junk ring J but it may be a separate piece. The advantage of this arrangement is that the packing rings may be examined or renewed without removing the piston itself.

It has been found that with high pressures the steam leaks behind

the packing rings and pressing them outwards a too rapid wear of the cylinder may be the result. It has therefore become a practice to place a limit to the expansion of the packing rings. One way of doing this is shown in Fig. 256. The solid distance ring G has a projecting part H lying between the packing rings D; the part H has recesses on top and bottom which receive corresponding projections on the packing rings. The rings are shown fully expanded. The same illustration shows the rings with circumferential grooves which hinder the leakage of steam past the rings. These grooves also retain the lubricating oil, but they should stop before reaching the ends of the rings as shown in the upper part of Fig. 251.

For large pistons there are many designs of packing in which there are two rings placed in the space between the junk ring and the piston flange, these rings being pressed outwards against the cylinder, and away from one another, one against the junk ring and the other against the piston flange, by one or more springs placed inside the rings. Fig. 257 shows one well known type of this packing (Buckley's). The spring is first made as a straight helical spring and then bent round and its ends united. The spring acting on the inclined surfaces of the rings D presses them outwards and one against the junk ring J and the other against the piston flange F. In a later design of the Buckley packing two springs are used in the manner shown in Fig. 258.

To prevent leakage past the packing rings the pressure of the rings against the cylinder need not exceed 2 lb. per square inch of bearing surface.

185. Stuffing-Boxes.—A *stuffing-box* is used where a sliding or rotating piece passes through the end or side of a vessel containing a fluid under pressure. The stuffing-box allows the sliding or rotating piece to move freely without allowing any leakage of the fluid. In the reciprocating steam engine the principal stuffing-boxes are where the piston rod passes through the end of the cylinder or cylinder cover and where the valve rod passes out of the valve chest.

An ordinary stuffing-box for a vertical rod is shown in Fig. 259. A is the rod and B a portion of the cylinder end, or cylinder cover, or valve chest end. C is the stuffing-box into the bottom of which is fitted the gun-metal neck bush D. The space between the rod and the wall of the stuffing-box contains the *packing* E of which there are many varieties. Apart from metallic packing, to be described later, ordinary packing may be simply greased hempen rope, or asbestos, or a combination of canvas or asbestos and a rubber core. Asbestos is more suitable for the higher temperatures than hemp.

The packing is compressed by screwing down the *gland* F by means of the nuts and studs H. When the gland is made of cast iron it is lined with a gun-metal bush K. Smaller glands are made of gun-metal without a bush. When the stuffing-box is vertical with the gland on top the latter may have an oil cup L formed on it, as shown.

A lubricating arrangement for an inverted gland is shown at (*a*) in Fig. 260. Oil is introduced into the annular recess M through the hole N. A shallow stuffing-box O contains soft cotton packing which becomes saturated with oil and forms a pad to lubricate the rod and prevents the oil running to waste down the rod.

At (*b*) in Fig. 260 is shown a gland for a horizontal rod. The flange of this gland contains an oil box and worsted siphon arrangement for delivering the oil slowly but regularly on to the rod.

A design of stuffing-box suitable for small rods is shown in Fig. 261. The gland F is moved into the stuffing-box by the nut T which works on a screw formed on the outside of the stuffing-box. All the parts of this stuffing-box would be made of gun-metal.

With high pressure, and consequently high temperature steam, ordinary soft packing for piston rod stuffing-boxes becomes charred by the heat, and some form of metallic packing is desirable. The ordinary stuffing-box in which the pressure of the packing on the rod is adjusted by tightening or slackening the gland by hand is unsatisfactory. The great danger is that the packing will be too tight and a considerable amount of power is then lost in friction.

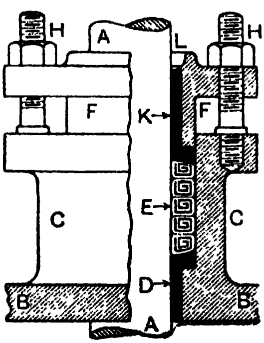


Fig. 259.

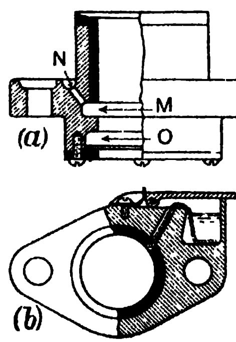


Fig. 260.

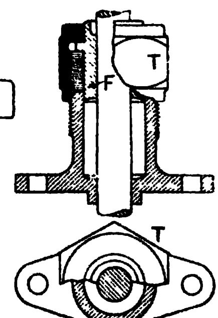


Fig. 261.

In an ideal stuffing-box the packing should be metallic and should be pressed against the rod by springs adjusted to give just sufficient pressure to prevent leakage. The only duty of the packing should be to prevent leakage; it should not serve as a bearing to support the rod laterally. There should also be provision for a slight lateral and a slight angular movement of the axis of the rod to allow for that axis not coinciding with the axis of the cylinder, an error which may be due to slightly faulty construction or to wear.

A well-known form of stuffing-box with metallic packing (Lancaster's) is shown in Fig. 262. The packing consists of two sets of blocks, each set consisting of four blocks, A, A, and B, B, the joints of the one set being at right angles to the joints of the other. The blocks A, A, are lined with white metal. The blocks are pressed against the rod by encircling springs D, each spring being made as a straight helical spring in two lengths which are joined together by a hook and eye at E, and by screwing the coils of one in between the coils of the other at F. By means of the connection at F the length of the spring and therefore the force which it exerts on the blocks may be adjusted.

The joint between the two rings H and K is spherical, and this permits of a slight angular movement of the piston rod. The piston

rod may also move laterally, carrying with it the packing blocks and rings. The rings and packing blocks are pressed together in the

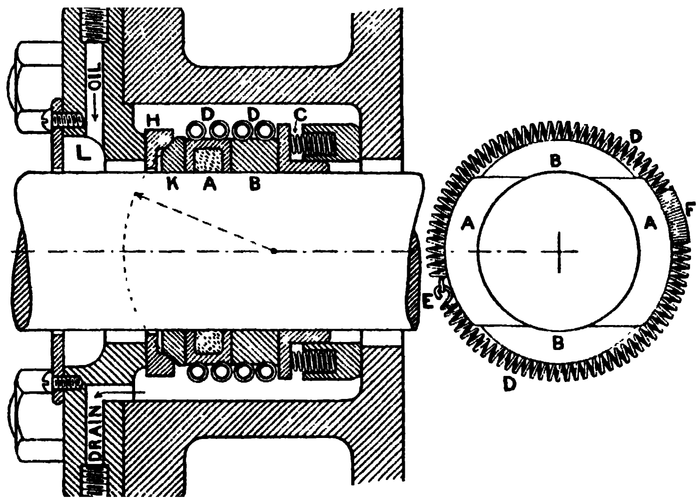


FIG. 262.—Lancaster's metallic packing.

direction of the axis of the piston rod by a number of springs C as shown. The annular space L may be filled loosely with soft cotton packing to retain the lubricating oil.

The United States metallic packing differs from that just described in that the blocks A, A and B, B are pressed directly against the rod by helical springs placed in recesses inside a casing which surrounds the blocks as shown in Fig. 263.

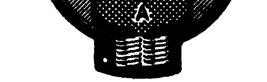


FIG. 263.

Dr. Schmidt, the inventor of Schmidt's superheater, has designed the stuffing-box and packing shown in Fig. 264. This stuffing-box is fitted with movable spherically seated packing rings A. The sleeve B containing the white metal packing rings C is cored out as shown to permit the access of air to prevent undue rise of temperature of the white metal rings when superheated steam is used.

The packing rings are held in position by the steam itself acting on the end of the bush D aided by the helical spring E. An oil box F containing loose soft packing is supplied with oil through the pipe H. It will be seen that the piston rod receives no lateral support from the stuffing-box or packing and this is, as has already been mentioned, as it should be. But in horizontal engines this involves the use of a tail rod to the piston unless the cylinder is allowed to give lateral support to the piston. The tail rod is supported either by a crosshead and guides outside the cylinder or by a sleeve attached to the cylinder

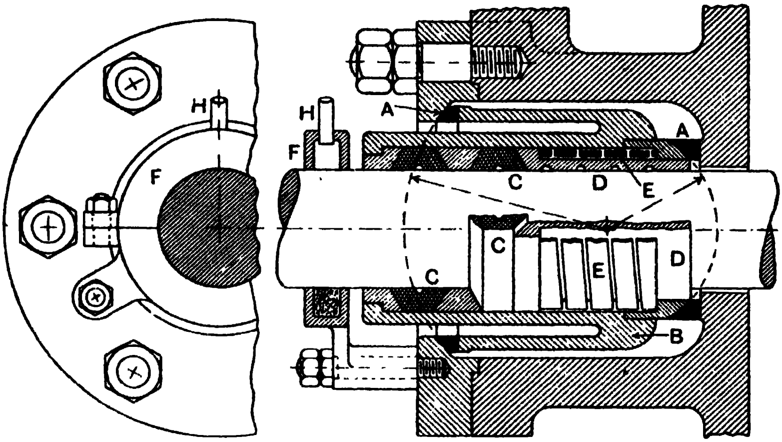


FIG. 264.—Schmidt's air-cooled stuffing-box and packing.

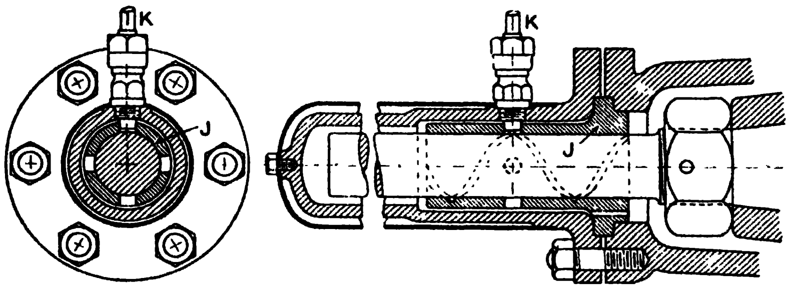


FIG. 265.—Schmidt's tail-rod guide

cover as shown in Fig. 265. The gun-metal bush J is lubricated with oil conveyed by the pipe K and distributed over the bearing surface by the helical groove shown.

A stuffing-box used for piston rods and valve spindles of large marine engines is shown in Fig. 266. This is fitted with three pairs of Beldam's metallic packing rings E. Each of these rings has an encircling spring and lies in a casing ring F. H, H are spring-loaded distance pieces. A pipe leading to the intermediate-pressure receiver is connected at K and another pipe leading to the low-pressure receiver is connected at L.

The stuffing-boxes used on the Allen high-speed engine illustrated on p. 237 are shown in Figs. 267 and 268. The first is for the valve

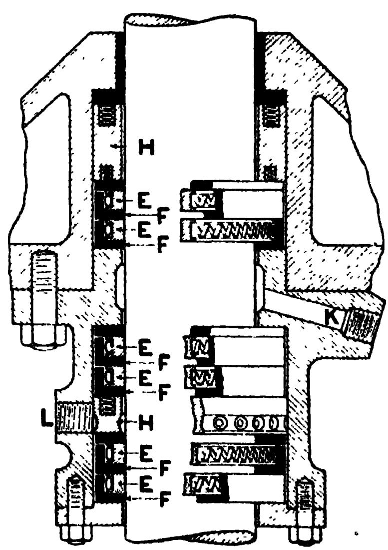


FIG. 266.

rods and the second for the piston rods. The white metal rings are lettered W. The bottom packing ring S for the valve rod is ordinary

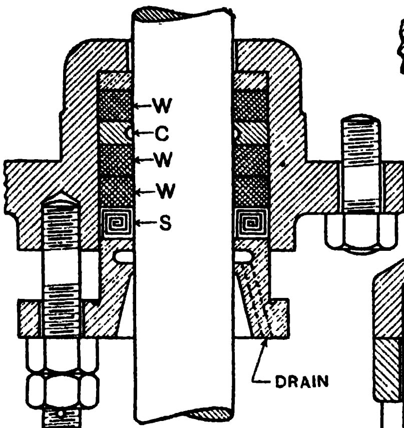


FIG. 267.—Valve rod stuffing-box.

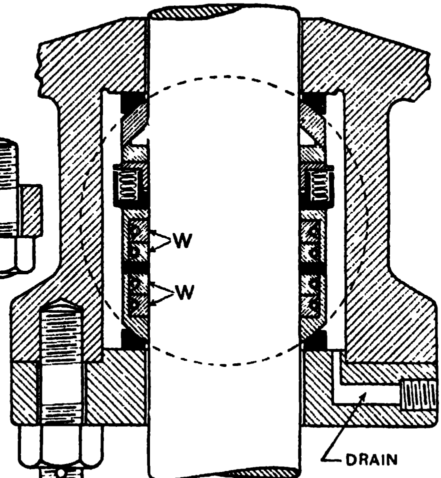


FIG. 268.—Piston rod stuffing-box.

packing. C is a cast iron ring. It will be seen that the piston rod packing permits of slight lateral and also slight angular movement of the rod.

One of the piston rod oil-scraping glands for the same engine is shown in Fig. 269. This is marked 14 in Fig. 223, p. 237.

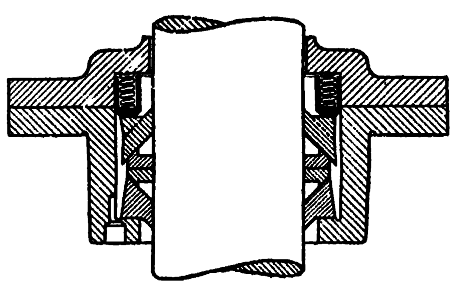


FIG. 269.—Piston rod oil-scraping gland.

186. Crossheads and Guides.—The outer end of the piston rod has forged on it or attached to it a *cross-head* to which is joined one end of the connecting rod, the other end of which embraces the crank pin. Owing to the obliquity of the connecting rod the forces acting along the piston rod and connecting rod require a third force to balance them and this third force is supplied by the guide upon which the crosshead slides. This is clearly shown by Figs. 270 and 271, where the forces acting on the crosshead are −P, the force acting along the piston rod; Q, the force acting along the connecting rod; and R, the reaction of the guide. The relative magnitudes of these forces are shown by the triangle of forces at the top in each illustration. The engine is supposed to be double-acting and horizontal in which case it will be seen that *R is an upward force during both the forward and return strokes* when the direction of rotation of the crank shaft is as

indicated. If the direction of rotation of the crank shaft be reversed then R becomes a downward force.

Crossheads vary greatly in form, but they may be roughly classified according to the number of slide or guide bars upon which they work. With one class four guide bars are used, with a second class there are two guide bars, and with a third class there is only one guide bar. In a fourth class the crosshead is attached to a piece called a slipper

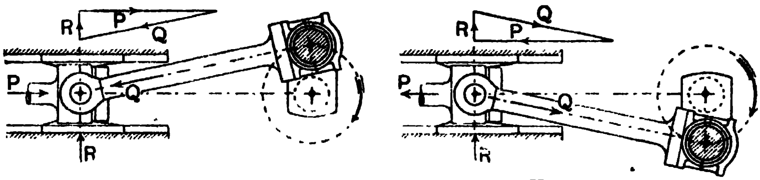


FIG. 270. FIG. 271.

which is guided by an arrangement equivalent to three guide bars. Differences in the design of crossheads also arise from arrangements for taking up the wear of the slide blocks and guides. The form of the crosshead also depends on the form of the crosshead pin and the method of fixing it to the crosshead or to the connecting rod which also involves the form of the connecting rod end.

Four-bar guides are now found chiefly on inside cylinder locomotives. In the example shown in Fig. 272 the forked crosshead C is forged on the end of the piston rod P. The crosshead pin or gudgeon G projects

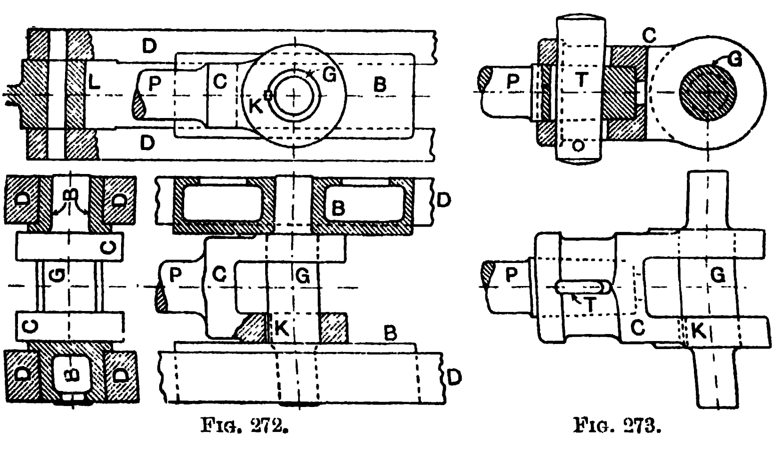


FIG. 272. FIG. 273.

through the prongs of the fork into cast iron or gun-metal slide blocks B which work each between a pair of wrought iron or steel guide bars D. The guide bars are bolted to lugs L cast on the side of the stuffing-box at one end and to a cross frame at the other. The end of the connecting rod comes in between the prongs of the fork of the crosshead. The crosshead pin is prevented from rotating by the key K.

When the crosshead used with a four-bar guide is not forged on the end of the piston rod it is generally a steel casting of the form shown in Fig. 273 and is secured to the piston rod by a cotter T. For the design shown in Fig. 273 the slide blocks would be the same as in Fig. 272. Frequently the cast steel crosshead has the slide blocks cast in one piece with it. Cast steel slide blocks should be faced with gun-metal or white metal so that the cast steel does not rub on the guide bars.

As an example of a crosshead for a slipper slide one of the crossheads of the Allen high-speed engine illustrated on p. 237 may be taken

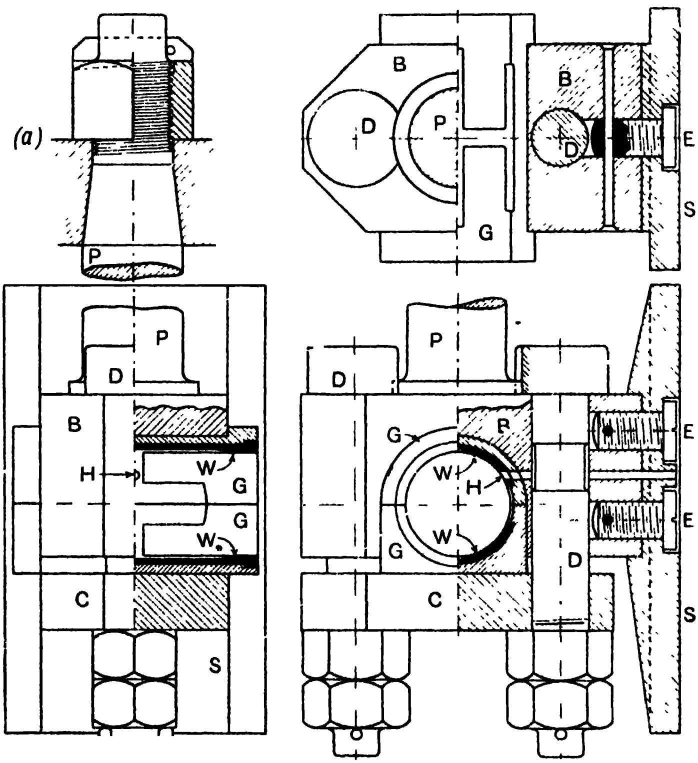


FIG. 274.—Crosshead of Allen engine.

and is shown in detail in Fig. 274. The piston rod P and the body B of the crosshead are of steel, forged in one piece. The two parts of the gun-metal bush G, which have a lining W of white metal, are held in the body B by means of the steel cap C and two bolts D. The slipper S, made of cast iron, is secured to the body of the crosshead by the screws E. The connection of the piston rod to the piston is shown at (*a*). From the illustration of the connecting rod of the same engine (Fig. 280, p. 261) it will be seen that the crosshead pin is fixed in the fork of the connecting rod. The same illustration (Fig. 280) also

shows how the lubricating oil reaches the pin under pressure. The oil is distributed between the pin and the bush G (Fig. 274) by means of the shallow recesses shown. A hole H through the top part of the bush G and through the body of the crosshead and through the slipper leads oil into a system of grooves on the sole of the slipper and lubricates the bearing surfaces of the slipper and guide.

A cross section of the guide is shown in Fig 275.

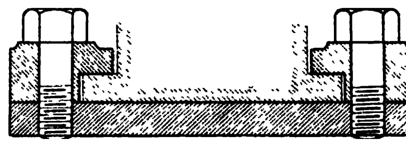


FIG. 275.

The slipper slide is not so suitable for an engine which has to run for any considerable time in the reverse direction on account of the smaller bearing surface of the upper part of the slipper on the guide.

An example of a crosshead for a two-bar guide, taken from American locomotive practice, is shown in Fig 276. C, the body of the crosshead, is a steel casting and is secured to the piston rod P by the cotter T. The gudgeon G is conical where it fits into the body of the crosshead. The slide blocks or shoes B are also steel castings but

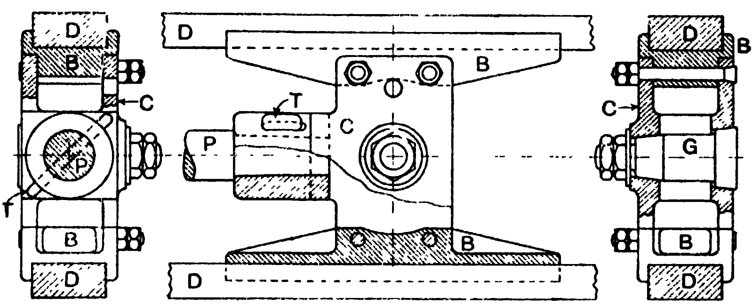


FIG. 276.

the surfaces in contact with the guide bars D are tinned to a depth of 1--16th of an inch.

A type of crosshead common in marine practice is shown in Fig. 277. C, the body of the crosshead, and G, the gudgeon, are of steel forged in one piece. The end of the piston rod P is partly conical and passes through the body of the crosshead to which it is secured by the nut N. Each guide block B is secured to the body of the crosshead by four bolts. The guides D are of cast iron and are bolted to the upright framing which supports the cylinder. The guide block to the right is the one which takes the pressure from the guide when the engine is running in the ahead direction and this is faced with white metal as shown. The other guide block is also frequently faced with white metal. The form of connecting rod end required for this type of crosshead is shown in Fig. 278.

In many stationary engines the guides have their bearing surfaces cylindrical, the axis of the cylindrical surface being the axis of the

piston rod. Such guides are bored and their ends are faced and recessed into the end of the engine cylinder in such a way that absolute alinement of the guides and piston rod is readily ensured.

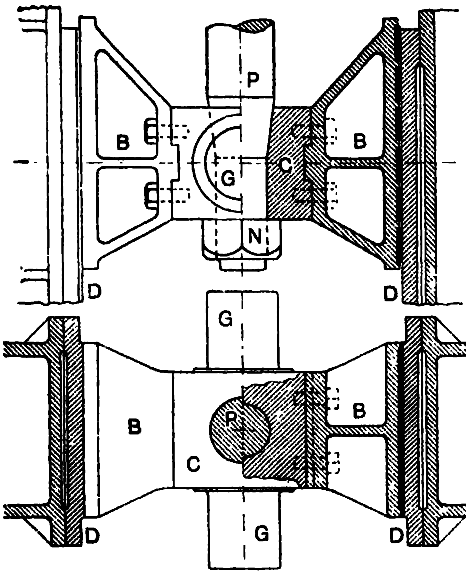


FIG. 277.

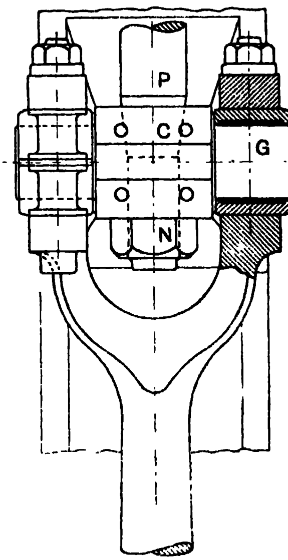


FIG. 278.

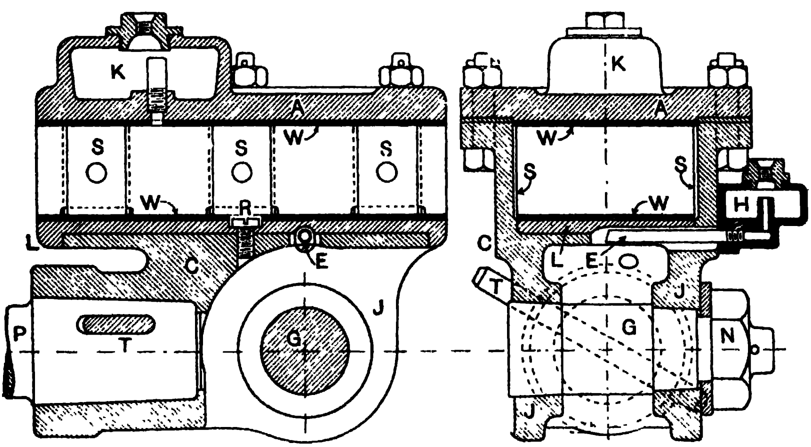
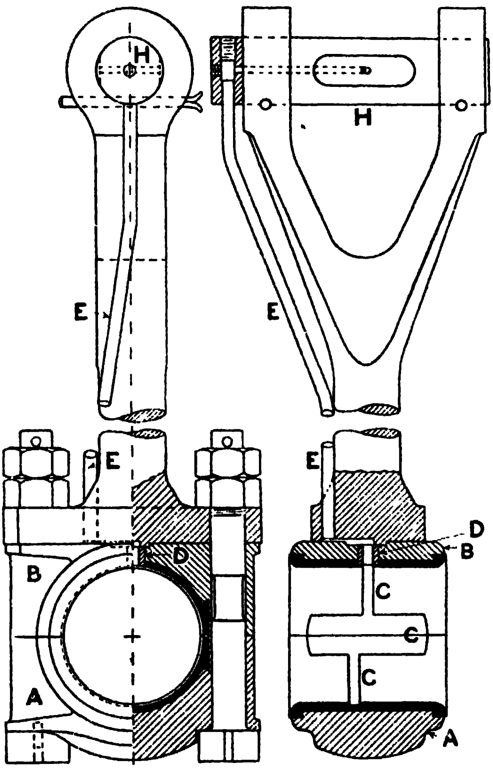


FIG. 279.—Crosshead for one-bar guide. (G.E.R.)

An example of a crosshead for a one-bar guide is shown in Fig. 279. This represents the practice on the Great Eastern Railway. C, the body of the crosshead is a steel casting which is secured to the piston rod P by the cotter T. The gudgeon G has conical bearings in the jaws J of the crosshead body and is secured by the nut N. The

liner L and the top part A are of cast iron and are faced with white metal W where they rub on the guide bar. The liner L is secured to the body of the crosshead by several screws of which one is shown at R. Phosphor bronze strips S are dovetailed into the crosshead body and provide the rubbing surfaces on the sides of the guide bar. The lubricator H supplies oil to the gudgeon bearing through the pipe E. The lubricator K supplies oil to the top of the guide bar through which there are holes at intervals to take the oil to the underside of the bar. The guide bar is made of wrought iron and is case hardened. One end of the guide bar is bolted to the cylinder while the other end is bolted to a transverse frame or "motion plate."

187. Connecting Rods.—The motion of the piston is transmitted through the piston rod to the crosshead and thence through the



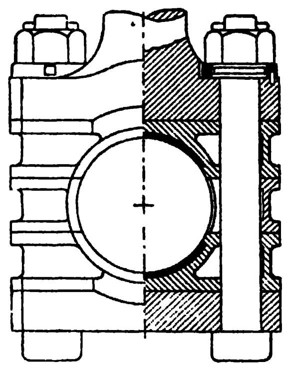


Fig. 281.

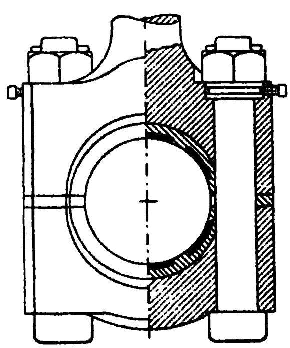


Fig. 280.

Fig. 282.

connecting rod to the crank pin. The connecting rod is generally a steel forging of circular, or rectangular, or **I**-section. At the crosshead end the gudgeon or crosshead pin is either fixed to the crosshead or to the connecting rod. In the former case the bearing for the pin is in the connecting rod while in the latter it is in the crosshead. At the other end the bearing is always in the connecting rod, the crank pin being fixed to or forged with the crank.

Various designs of connecting rods are shown in Figs. 280 to 285. In two of these (Figs. 280 and 283) the crosshead pin H is fixed to the connecting rod; in two of the others (Figs. 284 and 285) the crosshead pin would be fixed to the crosshead.

Referring to Fig. 280, this represents one of the connecting rods of of the Allen high-speed engine illustrated on p. 237. This is known as the marine type of connecting rod from the fact that practically all marine engines have this form of connecting rod. This type is however extensively used in all classes of reciprocating engines. The "brasses" A and B may be made of gun-metal but in the particular example shown in Fig 280 they are steel castings lined with white metal. When the brasses are made of gun-metal there is a separate steel cap between the bottom brass and the bolt heads as shown in Fig. 281. In the design shown in Fig. 282 the cap and the rod are forged together and after machining the cap is cut off. The design in Fig 282 is greatly superior to that in Fig. 281.

As explained on p. 236 a system of forced lubrication is used in the engine of which the connecting rod shown in Fig 280 is a part. Oil under pressure reaches the crank pin bearing through a hole in the crank shaft from an adjacent main bearing. The oil is distributed over the crank pin by the shallow recesses C in the white metal lining. Oil from the crank pin bearing passes through the gun-metal plug D, which is screwed into the brass B, and thence up through the tube E to the crosshead pin H.

The connecting rods shown in Figs. 283, 284, and 285 are from locomotive practice but are suitable for other types of horizontal

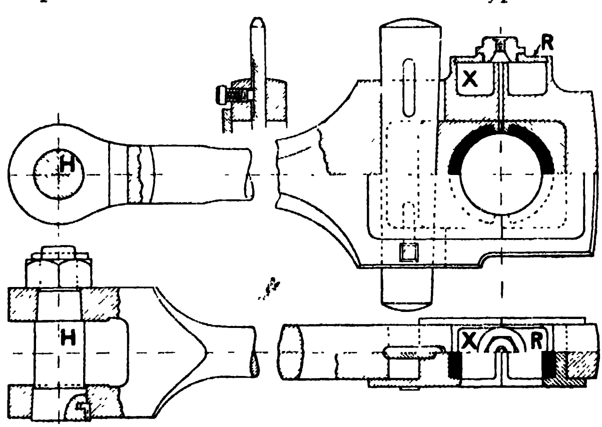


FIG. 283.—Locomotive connecting rod.

engines; they may also be used for vertical engines if the lubricating arrangements are altered. The design of the crank pin end of the rod in Fig. 283 is only suitable for an overhung crank as on an outside cylinder locomotive. It will be seen that the big end in Fig. 285 is a modified marine type. In the small end in Fig. 284 the bush is solid and there is no adjustment for wear there, but when necessary the bush may be replaced by a new one.

Note that in Fig. 285 there is adjustment for wear at both ends and, assuming that the wear is the same at both ends, the length of the rod between the centres of the bearings is not altered after adjusting for wear.

The oil boxes X are machined out of the solid. In Figs. 284 and 285 the oil boxes are cylindrical inside, but in Fig. 283 the interior of

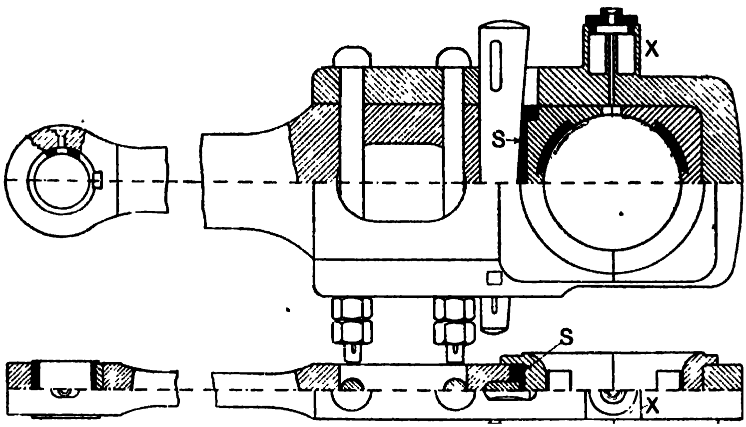


Fig. 284.—Locomotive connecting rod. (G.E.R.)

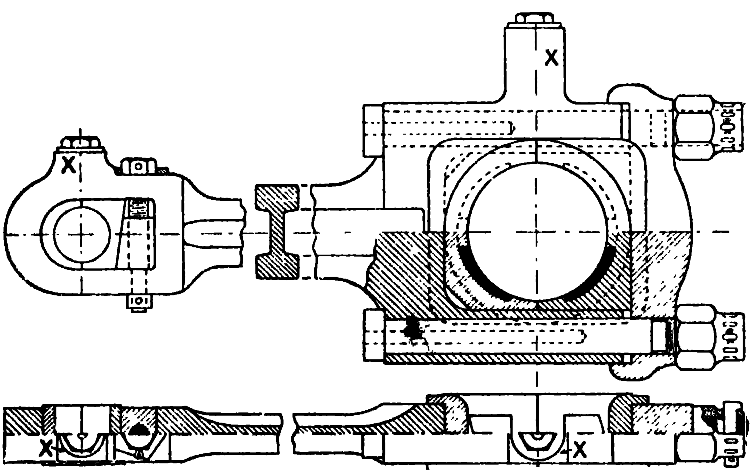


Fig. 285.—Locomotive connecting rod. (G.C.R.)

the box is rectangular and the cover R is bevelled on the edges and secured by riveting over the inner top edges of the box as shown.

Between the cotter and the inner brass of the big end in Fig. 284 there is a steel pad S to give a stronger bearing for the cotter.

188. Locomotive Coupling Rods.—One or more pairs of wheels on a locomotive may be coupled to the wheels on the driving axle in order

to increase the adhesion of the engine on the rails. The coupling rods used for this purpose are steel forgings and are either of rectangular or of **I**-cross section, generally the latter, and the cross section is the same throughout the whole length. The ends are now generally of the solid type and lined with solid gun-metal bushes without any adjustment for wear as shown in Fig 286.

When more than two pairs of wheels are coupled together the adjacent coupling rods are either joined on the same pin as shown in Fig. 287, or on an extra pin as shown in Fig. 288.

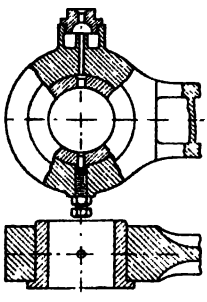


Fig. 286.

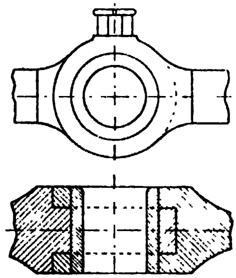


Fig. 287.

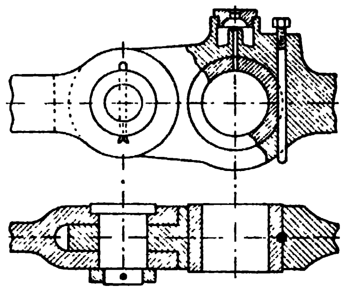


Fig. 288.

The cranks for the coupling rods on one side of an engine are at right angles to those on the other side.

189. Cranks and Crank Shafts.—A crank on the end of a shaft is called an overhung crank. An example of an *overhung crank* is shown in Fig. 289. The crank arm C and crank pin P are wrought iron or steel forgings. The crank pin is enlarged at D where it is enclosed in the crank arm, and the diameter of that part is about 1–500th part of the diameter larger than that of the hole in the crank which receives

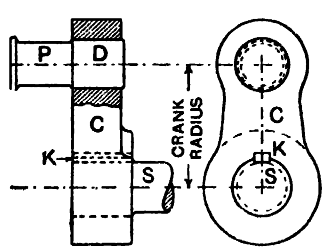


Fig. 289.—Overhung crank.

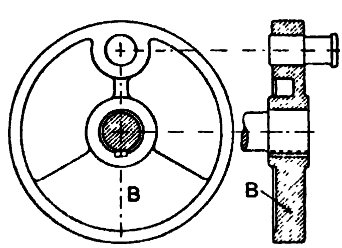


Fig. 290.

it. The crank is either heated and shrunk on to the pin or the pin is forced in by an hydraulic press. This is all that is necessary to secure the pin to the crank. The crank is secured to the shaft S in the same way, but it is usual to fit a key K in addition. The key shown is rectangular in cross section but round keys are also used.

A crank disc such as is shown in Fig. 290 sometimes takes the place of an ordinary forged crank. The disc is made of cast iron and is

recessed on the back except at B. The extra metal at B acts as a balance weight for the crank pin and a portion of the connecting rod. The pin and shaft are secured to the disc as already described for the forged crank.

The distance between the axes of the pin and shaft is called the *radius of the crank*, and when the axis of the piston rod produced intersects the axis of the shaft, which it does in most engines, the radius of the crank is half the stroke of the piston.

When the crank shaft has bearings on both sides of the crank the

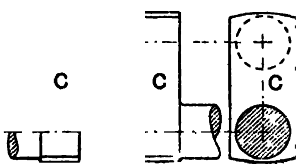


FIG. 291.

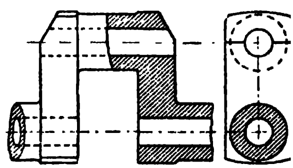


FIG. 292

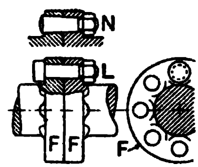


FIG. 293.

latter must have two arms with the crank pin between them. If the arms, pin, and shaft are in one forging, the shaft may be called a *cranked shaft*. A common form of cranked shaft is shown in Fig. 291. The arms or webs C are rectangular in cross section.

For the same strength a lighter cranked shaft is obtained by making the shaft and pin hollow as shown in Fig. 292, which represents the practice on warships having reciprocating engines.

When a crank shaft is not in one piece the different lengths are connected by couplings of the type shown in Fig. 293. The flanges F are forged on the ends of the parts of the shaft and are bolted together. The bolts may be parallel with round heads as shown at L in Fig. 293,

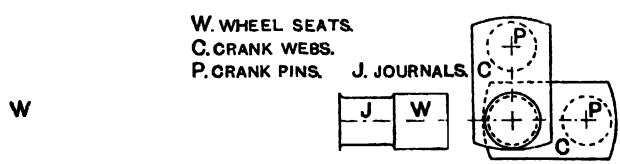


FIG. 294.—Locomotive cranked axle.

or they may be tapered and without heads as shown at N, or they may be tapered and have round heads as well.

Fig. 294 shows a cranked axle for a locomotive having inside cylinders. In locomotives with outside cylinders the cranks are overhung and are formed in the driving wheels.

Cranked shafts are steel forgings and when these assume large dimensions it is considered safer to make the shaft, crank arms, and crank pin separate forgings, which, after being machined, are put together to form a *built-up crank shaft*.

Fig. 295 shows a common form of built-up crank shaft. The arms are secured to the pin and shaft by shrinking or forcing as already described. Rectangular keys, or more commonly round pins are

driven into keyways partly in the shaft and partly in the crank arms as shown.

The built-up crank shaft is much heavier than a forged cranked shaft of the same strength. A crank shaft intermediate in weight to the built-up crank shaft and the forged cranked shaft is obtained by forging the crank pin and crank arms in one piece and fixing this to the shaft by shrinking on and keying as shown in Fig. 296.

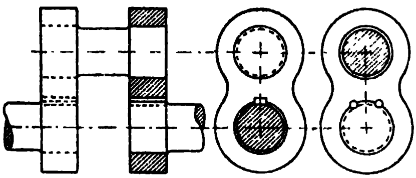


FIG. 295.

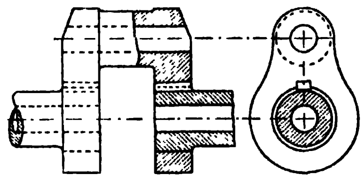


FIG. 296.

190. Eccentrics.—The eccentric is a modification of the crank, the difference being that in the eccentric the part corresponding to the crank pin is so large in diameter that it embraces the shaft. The evolution of the eccentric from the crank is shown in Figs. 297, 298, and 299. The crank in Fig. 297 has a pin P of the same diameter as the shaft S. In Fig. 298 the crank pin is larger in diameter than the shaft. Enlarging the crank pin until it embraces the shaft produces the eccentric, one form of which is shown in Fig. 299.

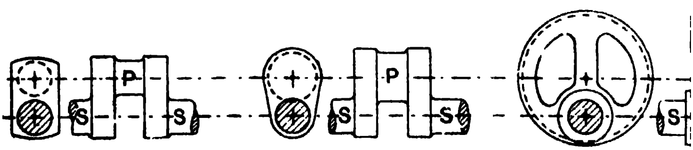


FIG. 297. FIG. 298. FIG. 299.

Kinematically the eccentric is identical with the crank and in studying the reciprocating motion given to a piece by means of an eccentric the eccentric is treated exactly as a crank having a radius equal to the distance from the axis of the eccentric disc or pulley or sheave to the axis of the shaft.

The eccentric is used for converting the rotary motion of a shaft into the reciprocating motion of a piece such as a slide valve. For this purpose the eccentric has an advantage over a crank in that no break or gap is needed in the shaft such as is necessary when a shaft is cranked at an intermediate point of its length.

Fig. 300 shows one of the eccentrics of the Allen high-speed engine illustrated on p. 237. The eccentric sheave is in two parts, A and B, to permit of it being put on the crank shaft between the cranks. The part A is made of cast iron and the part B is a steel forging. The two parts are fastened together by two bolts as shown. The eccentric strap C is in two parts bolted together. Both parts of the strap are steel castings. The strap is lined with white metal shown solid black in the sections. The lower end of the eccentric rod D is secured to

the upper part of the eccentric strap by two stud bolts. The upper end of D is forked to take the pin E which passes through the lower end of the valve rod F.

As explained on p. 236 a system of forced lubrication is used in the engine of which this eccentric is a part. Oil under pressure reaches the eccentric through a hole in the crank shaft from an adjacent main

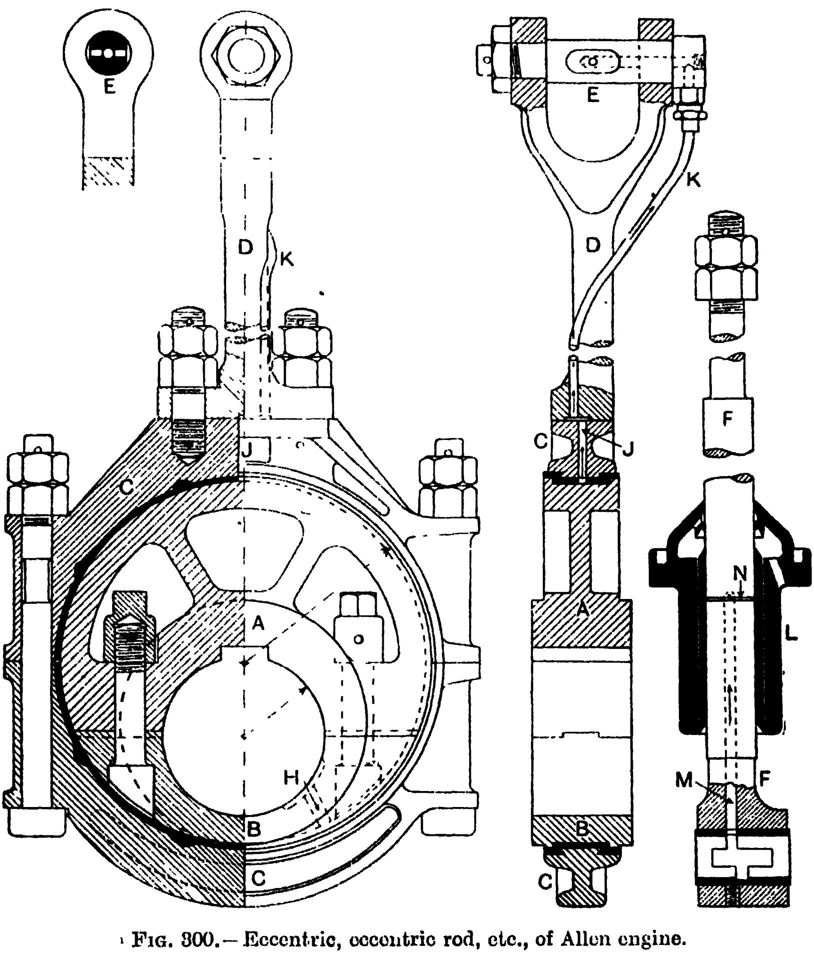


FIG. 300.—Eccentric, eccentric rod, etc., of Allen engine.

bearing. The oil enters the sheave at H and, passing between the bearing surfaces of the sheave and strap, goes through the hole J in the strap and up through the tube K to the pin E. The valve rod guide is lubricated by oil from the pin E through the hole M in the valve rod F to the groove N. The upper part of the oil gland L is so constructed as to wipe the oil off the valve rod and prevent it going up outside the trunk and crank chamber, also water from above

is wiped off and prevented from coming downwards into the trunk and crank chamber.

Fig. 301 shows an eccentric from locomotive practice. In this example the sheave and strap are made of cast iron and no white metal lining is necessary. There being no forced lubrication, an oil box for ordinary lubrication is provided.

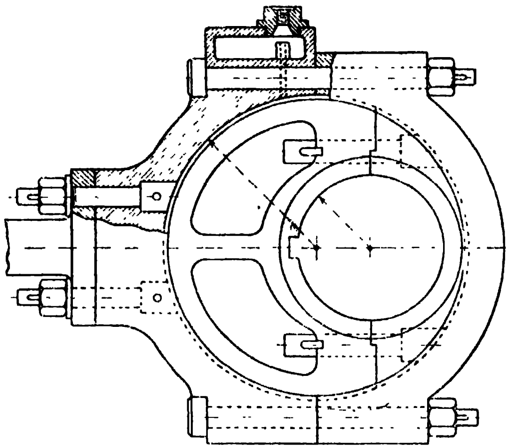


FIG. 301.—Locomotive eccentric.

191. Throttle Valves.—Steam is said to be throttled when the pipe or passage through which it is passing is contracted, and a valve used for this purpose is called a *throttle valve.* An ordinary stop valve may be used to throttle the steam on its way from the boiler to the engine by reducing the opening of the valve by hand. The throttle valve on an engine is however a special valve, placed as near to the engine as possible, and is operated by the governor.

FIG. 302.—Throttle valve.

The principal effect of throttling is a reduction of the pressure of the steam and therefore a reduction of the amount of work obtainable from the same volume of it. Throttling has also a drying or superheating effect on the steam.

A throttle valve operated by a governor must be of the balanced type and should require a very small effort to move it in any position.

An early form of throttle valve is shown in Fig. 302. This

consists of an elliptical plate placed in the steam pipe and mounted on a spindle which passes through a stuffing-box on the side of the pipe as shown. Although the valve is elliptical its edge is cylindrical and fits the pipe when in the closed position. The valve is operated by the governor through rods and levers connecting the sleeve of the governor and the valve spindle. When this type of valve is open its obliquity to the current of steam causes the resultant pressure on it to be

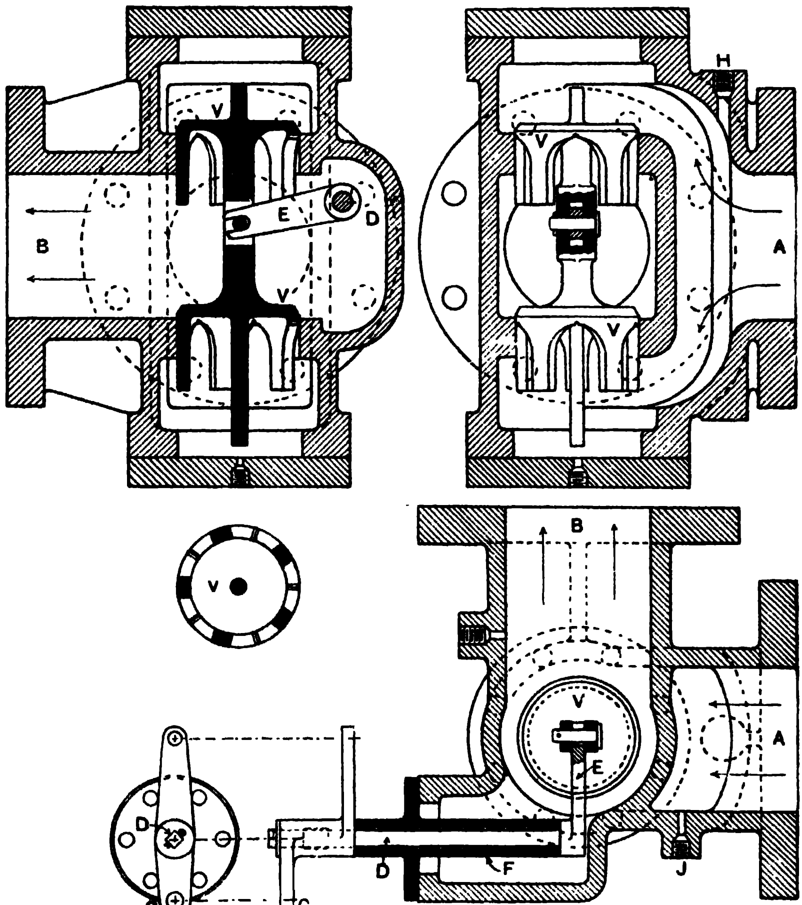


FIG. 303.—Throttle valve of Allen high-speed engine.

eccentric to the valve, being nearer to the edge which is up stream. This valve is therefore not quite balanced and the current of steam tends to close it.

Modern throttle valves are generally either of the piston type or of the double beat type. The throttle valve of the Allen high-speed engine illustrated on p. 237 is shown in detail in Fig. 303. The valve V is of the piston type. Steam from the boiler enters the

throttle valve chest at A and passes over to the top and to the bottom of the valve. When the valve is open the steam passes through the arch-shaped openings in the valve at top and bottom to the branch B and thence to the valve chest of the high-pressure cylinder. The valve is operated by the governor through the lever C, spindle D, and lever E. The spindle D has a long bearing in the sleeve F in which it is a good fit and no packing is required to keep it steam tight. A thermometer pocket is screwed in at H when superheated steam is used, otherwise a screwed plug is inserted. The screwed hole J is for a pressure gauge union.

Valves of the double beat type are liable to two defects. One is that the valve and its two seats, being separate castings, are liable to expand unequally when heated by the steam, with the result that although the valve may fit both seats properly when cold it may not do so when hot. When properly designed, however, and all the parts are made of the same material, this defect may be obviated. The other defect is that the valve is not quite balanced on account of the fact that one part of the valve has to pass through or over the seat of the other to get into position and one seat is therefore larger in diameter than the other. This defect is minimized by having the minimum amount of bearing surface on the seats.

A good design of double beat throttle valve is shown in Fig. 304. It will be seen that the effective diameters of the upper and lower seats are practically identical and the valve is therefore balanced.

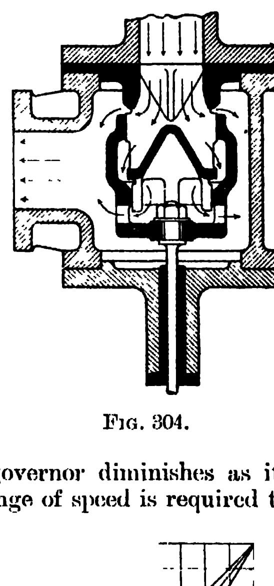


FIG. 304.

In a properly designed throttle valve, to be operated by a governor, the area of opening through the valve should diminish more rapidly as the closed position is approached. There are two reasons for this. In the first place the sensitiveness of the governor diminishes as its speed increases, that is to say, a greater change of speed is required to produce the same motion of the sleeve (see Fig. 385, p. 322). In the second place, as the opening of the throttle valve diminishes, the difference between the pressures on the two sides of the valve increases and therefore the velocity of the steam through the valve also increases.

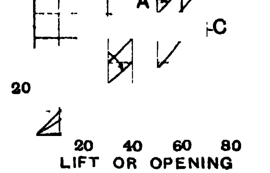


FIG. 305.

In Fig. 305 the area of opening of the valve, as a percentage of its full area of opening, and the lift or opening, as a percentage of its full lift or full opening, are plotted for three types of throttle valve. The straight line A applies to valves in which the area of opening is directly proportional

to the lift. This is the case with the double beat valve shown in Fig. 304 and also with piston valves in which the cutting off edges of the valve are parallel to the corresponding edges of the ports over which the valves travel. The curve B applies to the older form of throttle valve shown in Fig. 302. The curve C applies to the Allen throttle valve shown in Fig. 303. The very gradual closing in the case of the Allen valve is due to the shape of the openings in the pistons which it will be seen finish in small inverted V-notches.

192. Locomotive Throttle Valve or Regulator.—The valve which controls the steam supply from the locomotive boiler to the steam pipe which leads to the cylinder valve chests corresponds to the *stop valve* in other steam engines. In America this valve is called the *throttle valve*, but in British practice it is known as the *regulator valve* or simply

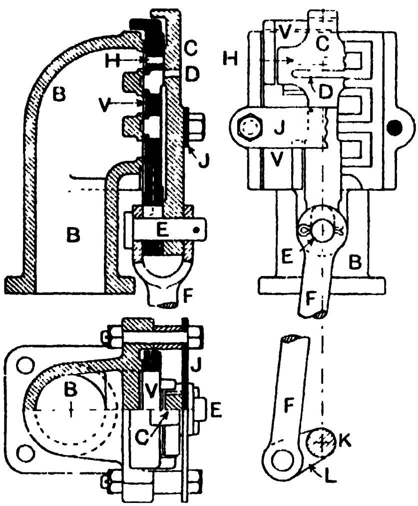


FIG. 306.

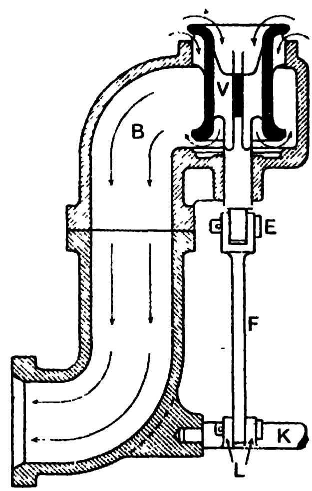


FIG. 307.

as the *regulator*. There are two main types of regulator in use, the slide valve type and the double beat type.

An example of the slide valve type of regulator is shown in Fig. 306. V is a slide valve which travels over three rectangular ports in the face of the chest or body B on the top of a vertical pipe in the steam dome. On the back of the valve V there is a supplementary slide valve C having one small port D. A pin E in the end of the link F passes through a slotted hole in the main valve V and through a fitting hole in the supplementary valve C. When the link F is raised the valve C moves, but the valve V remains at rest until the port D in the supplementary valve comes opposite to the port H in the main valve, when the two valves move together. The area of the supplementary valve being small this valve is easily opened, and once open, the steam reaches both sides of the main valve, which is then nearly balanced and a comparatively small force will then be sufficient

to open it. The valves are kept on their seats by a flat spring plate J. The valve is operated from the cab through the shaft K, crank L, and link F.

An example of the double beat type of regulator is shown in Fig. 307. The valve V has two seats the upper of which is larger than the lower, for an obvious reason, and in consequence the valve is not quite balanced. The valve is operated in the same way as already described for the slide valve type of regulator.

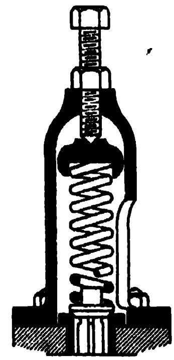


FIG. 308.

193. Relief Valves.—The ends of steam cylinders are generally provided with *relief* or *escape valves* which are spring loaded. The main purpose of these valves is to prevent excessive pressure at the end of the piston stroke due to the presence of water formed by the condensation of steam. A common form of relief valve is shown in Fig. 308.

CHAPTER XIV

HYPOTHETICAL INDICATOR DIAGRAM PROBLEMS

194. Hypothetical Indicator Diagram.—Fig. 309 is a diagram which shows the variation of the steam pressure on one side of the piston of a steam engine during two consecutive strokes.. KL represents the stroke of the piston and also the volume swept through by the piston in one stroke. The distance OK represents the clearance volume, that is, the volume of the space between the admission valve and the piston when the left hand face of the piston is at K, the beginning of the forward stroke.

Consider only what takes place on the left hand side of the piston. At the beginning of the forward stroke the steam pressure on the piston is represented by KA. The horizontal line AB indicates that the pressure on the piston is constant while the piston moves from K to H. When the piston is at H the admission of steam to the cylinder is suspended and the volume of steam in the cylinder to the left of the piston is represented by OH. As the piston moves forward from H the steam expands, its volume increasing from OH at the point of cut-off to OL at the end of the forward stroke, and its pressure falls from HB to LC. At L the exhaust valve opens and the steam begins to escape, say, into the atmosphere, and its pressure at once falls from LC to LD. The piston now begins its return stroke and until it reaches the point N the steam pressure remains constant as shown by the horizontal line DE. When the piston is at N the exhaust valve closes and the steam remaining in the cylinder is compressed while the piston completes its return stroke, the volume of the steam being reduced to OK, the clearance volume, and its pressure raised from NE to KF. The piston having now completed its return stroke the admission valve opens and the steam pressure in the cylinder at once rises to KA and the next forward stroke begins.

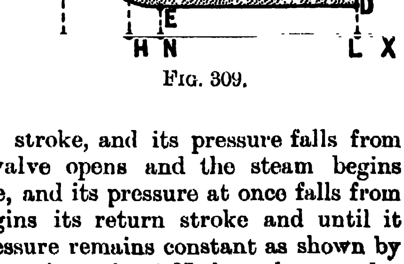


FIG. 309.

The curves BC and EF, whose co-ordinates show the relation between the pressure and volume of the steam while it expands or is compressed, are assumed to be rectangular hyperbolas of which OX and OY are the rectangular axes; hence, $pv =$ constant.

The area KABCL represents the work done *by* the steam on the left hand face of the piston during the forward stroke, and the area LDEFK represents the work done *against* the steam on the same side of the piston during the return stroke. The *effective* work done on the left hand side of the piston during the two strokes considered is therefore represented by the shaded area FABCDE.

Assuming that the diagram of work done on the right hand side of the piston is the same as that shown in Fig. 309, except that it will be turned over from left to right, the total effective work done on the piston during one revolution of the crank shaft will be represented by *twice* the shaded area, or the shaded area will represent the total effective work done on the piston during one stroke.

195. Mean Effective Pressure on Piston.—Taking the hypothetical diagram of work described in the preceding Art., a formula will now be obtained for finding the mean height of the diagram in terms of the quantities p_1, r, c, p_0, and x marked on the diagram in Fig. 310. l denotes the length of the stroke of the piston or the volume swept through by the piston in one stroke; c is the fraction which the clearance volume is of the volume swept through by the piston in one stroke; r is the fraction which the distance travelled by the piston during admission is of the stroke; x is the fraction which the distance travelled by the piston during compression is of the stroke; p_1 is the absolute initial pressure of the steam, and p_0 is the absolute back pressure during exhaust.

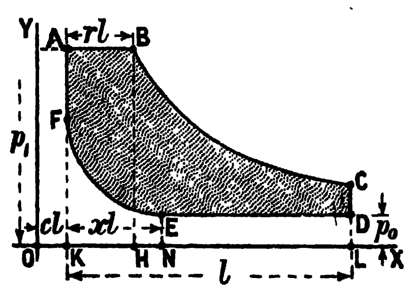


FIG. 310.

In what follows the formula proved in Art. 36, p. 35, is used for the areas of the figures BCLH and EFKN under the hyperbolas BC and EF.

Let p_m denote the mean height of the shaded area ABCDEF in units of pressure.

Area ABCDEF

$$= \text{area ABHK} + \text{area BCLH} - \text{area DENL} - \text{area EFKN}$$

$$= \text{AK} \cdot \text{KH} + \text{BH} \cdot \text{OH} \log_e \frac{\text{OL}}{\text{OH}} - \text{DL} \cdot \text{LN} - \text{EN} \cdot \text{ON} \log_e \frac{\text{ON}}{\text{OK}}$$

$$p_m l = p_1 rl + p_1(rl + cl) \log_e \frac{1+c}{r+c} - p_0(l - xl) - p_0(xl + cl) \log_e \frac{x+c}{c}$$

Dividing both sides by l

$$p_m = p_1 r + p_1(r + c) \log_e \frac{1+c}{r+c} - p_0(1 - x) - p_0(x + c) \log_e \frac{x+c}{c}$$

or $$p_m = p_1 \left\{ r + (r + c) \log_e \frac{1+c}{r+c} \right\} - p_0 \left\{ 1 - x + (x + c) \log_e \frac{x+c}{c} \right\}$$

If there is no compression, $x = 0$, then

$$p_m = p_1\left\{r + (r + c)\log_e \frac{1 + c}{r + c}\right\} - p_0$$

If there is no compression and no clearance, $x = 0$, and $c = 0$, then

$$p_m = p_1 r\left(1 + \log_e \frac{1}{r}\right) - p_0$$

The logarithms used in the above formulæ are Napierian or hyperbolic logarithms. The Napierian logarithm is obtained by multiplying the common logarithm by 2·3026.

If the diagrams for the two ends of the cylinder are alike and the faces of the piston have the same area then p_m is the *mean effective pressure* on the piston during one stroke.

196. Indicated Horse-Power.—Assuming that the faces of the piston have the same area and that the diagrams of work on the two faces are alike, then if p_m is in pounds per square inch, the indicated horse-power is given by the formula $H = \frac{2 \times 0{\cdot}7854 d^2 p_m l N}{33{,}000}$, where d = diameter of cylinder in inches, l = stroke of piston in feet, and N = revolutions of crank shaft per minute.

The above is for a single cylinder engine. Where an engine has more than one cylinder the horse-power of the engine is the sum of the horse-powers of the several cylinders.

197. Steam Consumption Calculated from Indicator Diagram.—Referring to the diagram Fig. 311, v, the volume swept through by the piston in one stroke, is represented by the length of the diagram. $v = \frac{0{\cdot}7854 d^2 l}{144}$ cubic feet, where d is the diameter of the cylinder in inches, and l is the stroke of the piston in feet.

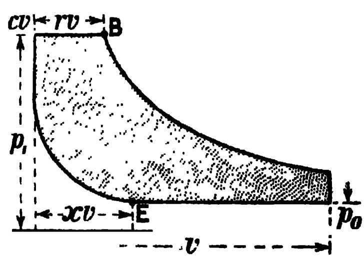


FIG. 311.

At the point of cut-off B the volume of steam in the cylinder to the left of the piston is $(r + c)v$ cubic feet, and the weight of this steam is $(r + c)vw_1$, where w_1 is the weight of one cubic foot of steam at pressure p_1. While the piston travels from the point of cut-off to the end of the forward stroke and back again as far as the point E, at which compression begins, no more steam enters the cylinder to the left of the piston, and at E the weight of steam remaining in the cylinder to the left of the piston is $(x + c)vw_0$, where w_0 is the weight of one cubic foot of steam at pressure p_0. The weight of exhaust steam which has left the cylinder to the left of the piston is evidently $(r + c)vw_1 - (x + c)vw_0$ and this is the amount of steam used in the cylinder to the left of the piston during one revolution of the crank shaft. If the diagrams for opposite sides of the piston are alike then the total weight of steam used per revolution is twice the above amount or $2\,v\{(r + c)w_1 - (x + c)w_0\}$.

The steam consumption determined as above is called the *indicated steam* and this is generally considerably less than the actual weight of steam used because no account has been taken of the steam condensed in the cylinder.

198. Effect of Clearance, Expansion, and Compression on Economy.—The student will find it instructive to study the results of the following calculations on the performance of a steam engine working under hypothetical conditions. The object of the calculations is to show the effect of clearance, expansion, and compression on the economy of an engine.

The engine has a single cylinder and is double acting. Diameter of cylinder, 10 inches. Stroke of piston, 18 inches. Revolutions per minute, 120. Initial pressure of steam, 80 lb. per sq. inch absolute. Back pressure, 16 lb. per sq. inch absolute. The expansion and compression curves of the indicator diagram will be assumed to be hyperbolic. The steam admission will be assumed to begin at the beginning of the stroke and the release at the end of the stroke. The steam consumption to be calculated will be the *indicated steam.*

In the formula for the horse-power the only variable will be the mean effective pressure, hence,

$$\text{horse-power} = \text{H} = \frac{0{\cdot}7854 \times 10^2 p_m \times 1{\cdot}5 \times 2 \times 120}{33{,}000} = 0{\cdot}857 p_m$$

The formula for p_m is,

$$p_m = 80\left\{r + (r + c)\log_e \frac{1 + c}{r + c}\right\} - 16\left\{1 - x + (x + c)\log_e \frac{x + c}{c}\right\}$$

The volume swept through by the piston in one stroke is,

$$v = \frac{0{\cdot}7854 \times 10^2 \times 1{\cdot}5}{144} = 0{\cdot}818 \text{ cubic foot.}$$

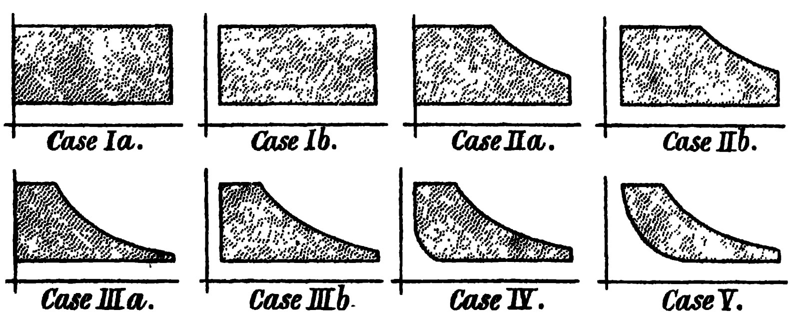


FIG. 312.

Taking the density of steam at 80 lb. pressure as 0·183 lb. per cubic foot, and the density of steam at 16 lb. pressure as 0·040 lb. per cubic foot, the expression for the steam used per revolution is

$$2 \times 0{\cdot}818\{0{\cdot}183(r + c) - 0{\cdot}04(x + c)\}$$

and the steam used per hour in pounds is,

$$\begin{aligned} \text{W} &= 120 \times 60 \times 2 \times 0{\cdot}818\{0{\cdot}183(r + c) - 0{\cdot}04(x + c)\} \\ &= 11{,}779(0{\cdot}183r + 0{\cdot}143c - 0{\cdot}04x). \end{aligned}$$

The hypothetical indicator diagrams of the various cases to be considered are shown in Fig. 312, and the particulars as regards clearance, expansion, and compression, and the results of the calculations are given in the table below.

W/H is the weight of steam used in pounds per horse-power per hour.

Case.	c	r	x	p_m	H	W	W/H
I*a*	0·0	1·0	0·0	64·0	54·8	2156	39·3
I*b*	0·1	1·0	0·0	64·0	54·8	2324	42·4
II*a*	0·0	0·5	0·0	51·7	44·3	1078	24·3
II*b*	0·1	0·5	0·0	53·1	45·5	1246	27·4
III*a*	0·0	0·25	0·0	31·7	27·2	539	19·8
III*b*	0·1	0·25	0·0	36·1	30·9	707	22·9
IV	0·1	0·25	0·15	34·8	29·8	637	21·4
V	0·1	0·25	0·40	29·6	25·4	519	20·4

The relative economy of the engine in the different cases, so far as steam consumption is concerned, is shown by the numbers in the last column of the table, W/H being the weight of steam required to give one horse-power for one hour.

An examination of the table shows that one effect of clearance is to increase W/H and on this account clearance is objectionable. It will be observed however that the addition of clearance increases the power of the engine, which is an advantage, but the increase in power so obtained is accompanied by a steam consumption much greater in proportion. For example, comparing cases III*a* and III*b*, the addition of clearance has here the effect of increasing the power by 3·7 horse-power, but the additional steam used amounts to 168 lb. per hour or 45·4 lb. per hour for each horse-power added due to clearance.

The effect of expansion on economy is very marked but it must be observed that the increase in economy is accompanied by a decided diminution in the power of the engine, so that with an increase of expansion a larger engine would be required to develop the same power, the other conditions remaining the same. For example, comparing cases II*b* and III*b*, the engine considered, which has a cylinder 10 inches in diameter and a piston stroke of 18 inches, develops 45·5 horse-power in case II*b*. To develop this power under the conditions of case III*b* would require a cylinder 11·5 inches in diameter with a piston stroke of 20 inches.

The effect of compression is clearly to neutralize partly the bad effect of clearance on the economy. But this advantage of compression is accompanied by a diminution in power necessitating a larger engine to develop the same power.

199. Relation between Steam Consumption and Power—Willans Line.—Using the notation and formulæ of the preceding Arts., the steam consumption per hour is,

$$W = 2N \times 60v\{(r + c)w_1 - (x + c)w\}$$

Now, over a considerable range of pressure the relation between the density of the steam and the pressure is very approximately a linear one and may be written, $w = ap + b$, where a and b are constants. For example, over the range of pressure between 10 lb. per square inch and 120 lb. per square inch the weight of a cubic foot of steam, in pounds, is given very approximately by the formula, $w = 0{\cdot}0022p + 0{\cdot}006$.

Hence, $W = 120Nv\{(r + c)(ap_1 + b) - (x + c)(ap_0 + b)\}$.

Consider the case of an engine which is governed by varying the initial pressure p_1. The quantities v, r, c, x, and p_0 are constant and if the speed N is also assumed to be constant, the formula for W reduces to $W = k_1p_1 - k_2$ where k_1 and k_2 are constants.

Taking the engine whose performance was studied in Art. 198, $v = 0{\cdot}818$, $N = 120$, $p_0 = 16$, and $c = 0{\cdot}1$. Now take $r = 0{\cdot}5$, and $x = 0{\cdot}15$, and use the formula given above for the density of the steam, namely, $w = 0{\cdot}0022p + 0{\cdot}006$. Then $W = 15{\cdot}55p_1 - 78{\cdot}9$.

The formula for the mean effective pressure is,

$$p_m = p_1\left\{r + (r + c)\log_e\frac{1 + c}{r + c}\right\} - p_0\left\{1 - x + (x + c)\log_e\frac{x + c}{c}\right\}$$

If r, c, x, and p_0 are constant, then the formula for p_m reduces to $p_m = k_3p_1 - k_4$, where k_3 and k_4 are constants. For example, if $r = 0{\cdot}5$, $c = 0{\cdot}1$, $x = 0{\cdot}15$, and $p_0 = 16$, then $p_m = 0{\cdot}864p_1 - 17{\cdot}3$.

For a given engine running at a given speed the horse-power H is given by $H = k_5p_m = k_5k_3p_1 - k_5k_4 = k_6p_1 - k_7$, where k_6 and k_7 are constants.

From the equation $W = k_1p_1 - k_2$, $p_1 = \dfrac{W + k_2}{k_1}$

and from the equation $H = k_6p_1 - k_7$, $p_1 = \dfrac{H + k_7}{k_6}$

Hence $\dfrac{W + k_2}{k_1} = \dfrac{H + k_7}{k_6}$ or $W = \dfrac{k_1}{k_6}H + \dfrac{k_1k_7 - k_2k_6}{k_6}$

which is of the form $W = mH + k$, where m and k are constants. The relation between W and H is therefore a linear one.

Still considering the engine whose performance was studied in Art. 198, and taking $r = 0{\cdot}5$ and $x = 0{\cdot}15$, it was shown that $H = 0{\cdot}857p_m$, and it has been shown above that $p_m = 0{\cdot}864p_1 - 17{\cdot}3$, therefore $H = 0{\cdot}74p_1 - 14{\cdot}8$, and $p_1 = \dfrac{H + 14{\cdot}8}{0{\cdot}74}$. Also from $W = 15{\cdot}55p_1 - 78{\cdot}9, p_1 = \dfrac{W + 78{\cdot}9}{15{\cdot}55}$. Hence $\dfrac{W + 78{\cdot}9}{15{\cdot}55} = \dfrac{H + 14{\cdot}8}{0{\cdot}74}$, which reduces to $W = 21H + 232$.

The graph of $W = 21H + 232$ is shown in Fig. 313.

Altering r to $0{\cdot}25$, and keeping the other data the same the student should show that $W = 16{\cdot}3H + 144$. The graph of this is also shown in Fig. 313.

The straight lines whose ordinates and abscissæ give the relation between the steam consumption per hour and the horse-power when the speed and cut-off are constant are called *Willans Lines*, after the late

Mr. P. W. Willans who made numerous tests of steam engines. Mr. Willans found that in actual engines running at constant speed with constant cut-off the relation between the steam consumption per hour and the horse-power (whether indicated horse-power or brake horse-power) was approximately a linear one, the power being varied by varying the initial pressure.

The Willans lines which have been obtained in this Art. are theoretical Willans lines. Examples of actual Willans lines will be found on p. 391.

Dividing both sides of the equation $W = 21H + 232$ by H, then $\frac{W}{H} = 21 + \frac{232}{H}$ gives the relation between the steam consumption per horse-power per hour (W/H) and the horse-power (H) for the case where $r = 0{\cdot}5$. The graph of this is a curve A shown in Fig. 313.

In like manner for the case where $r = 0{\cdot}25$, $\frac{W}{H} = 16{\cdot}3 + \frac{144}{H}$. The graph of this is the curve B shown in Fig. 313.

Since the horse-power H at a given speed is proportional to the mean effective pressure p_m, the relation between W and p_m will also be a linear one when the speed and cut-off are constant. Again, weight of fuel required to produce a given weight of steam W may be assumed to be proportional to W. Hence the relation between fuel consumption per hour and horse-power is a linear one when the speed and cut-off are constant.

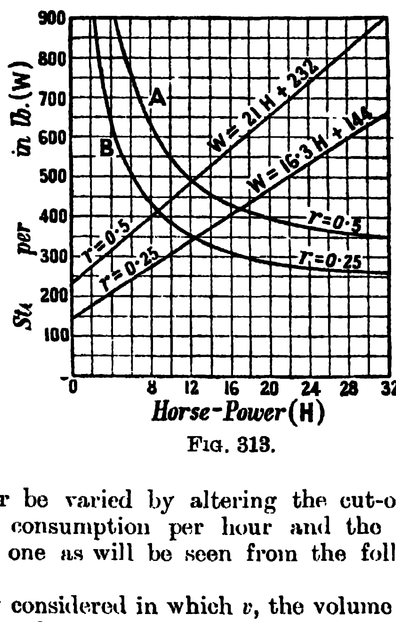


FIG. 313.

If the initial pressure be kept constant and the power be varied by altering the cut-off, the relation between the steam consumption per hour and the horse-power is no longer a linear one as will be seen from the following example.

Taking the engine already considered in which v, the volume swept through by the piston in one stroke, is 0·818 cubic foot. Let $N = 120$, $c = 0{\cdot}1$, $x = 0{\cdot}15$, $p_0 = 16$, and $p_1 = 100$. H and W are to be calculated for the following values of r, namely, 0·7, 0·6, 0·5, 0·4, 0·3, 0·2, and 0·1.

The formulæ required are

$$p_m = p_1\left\{r + (r + c)\log_e\frac{1 + c}{r + c}\right\} - p_0\left\{1 - x + (x + c)\log_e\frac{x + c}{c}\right\}$$

$$= 100\left\{r + (r + 0{\cdot}1)\log_e\frac{1{\cdot}1}{r + 0{\cdot}1}\right\} - 17{\cdot}3$$

$$H = 0{\cdot}857 p_m$$
$$W = 2N \times 60v\{(r + c)w_1 - (x + c)w_0\}$$
$$= 2 \times 120 \times 60 \times 0{\cdot}818\{(r + 0{\cdot}1) \times 0{\cdot}226 - 0{\cdot}25 \times 0{\cdot}04\}$$
$$= 2662r + 148{\cdot}4.$$

	0·7	0·6	0·5	0·4	0·3	0·2	0·1
p_m	78·2	74·3	69·1	62·1	53·2	41·7	26·8
H	67·0	63·7	59·2	53·2	45·6	35·7	23·0
W	2012	1746	1479	1213	947	681	415

The results are tabulated above and they have been plotted and joined by a fair curve in Fig. 314.

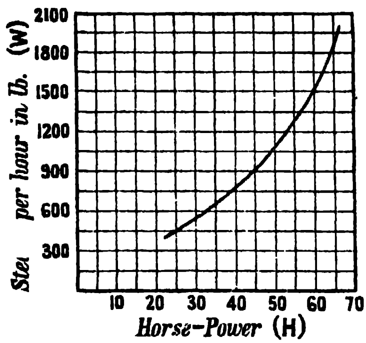


FIG. 314.

Exercises XIV

1. In an indicator diagram of the type shown in Fig. 310, p. 274, $r = 0{\cdot}35$, $c = 0{\cdot}08$, $x = 0{\cdot}25$, $p_1 = 75$ lb. per square inch, and $p_0 = 16$ lb. per square inch. Using the formula of Art. 195, calculate p_m the mean effective pressure in lb. per square inch.

Draw the diagram to the pressure scale of 1 inch to 20 lb. per square inch on a stroke base 5 inches long. Divide the diagram into 10 vertical strips of equal width, and at the middle of the width of each strip erect an ordinate. Find the mean pressure by taking the mean of these 10 mid ordinates.

If the diameter of the cylinder is 8 inches, stroke of piston 16 inches, and revolutions per minute 100, calculate the horse-power of the engine, using the mean pressure calculated by the formula.

Taking the densities of the steam of 75 and 16 lb. per square inch pressure as 0·172 and 0·04 lb. per cubic foot respectively, calculate the indicated steam in pounds per horse-power per hour.

2. In a steam engine cylinder the clearance volume is 5 per cent. of the volume swept through by the piston in one stroke, and the back pressure is 17 lb. per square inch. If compression begins at 0·3 of the stroke from the end of the exhaust stroke, find the pressure at the end of the compression. Also find where compression should begin in order that the pressure at the end of the compression may be 85 lb. per square inch. Assume hyperbolic compression.

3. The cylinder of a steam engine is 12 inches in diameter and the piston stroke is 22 inches. The mean effective steam pressure on the piston is 50 lb. per square inch, and the crank shaft makes 110 revolutions per minute. Determine the diameter of the cylinder of another engine, which, with a crank shaft speed of 100 revolutions per minute and a mean effective pressure of 40 lb. per square inch, will develop the same power. Assume the stroke of the piston of the second engine to be 1·8 times the diameter of the cylinder.

4. The cylinder of a steam engine is 12 inches in diameter and the stroke of the piston is 22 inches. The initial pressure of the steam is 85 lb. per square inch absolute and the back pressure is 17 lb. per square inch absolute. The pressure at the end of the compression is equal to the initial pressure. The steam is cut off at 0·3 of the stroke. The clearance is 7 per cent. of the volume swept through by the piston in one stroke. The crank shaft speed is 130 revolutions per minute. Calculate the horse-power, the weight of indicated steam per

hour, and the steam per horse-power hour. Assume hyperbolic expansion and compression, and take the density of steam at 85 lb. pressure as 0·194 lb. per cubic foot.

5. In a steam engine the diameter of the cylinder is 8 inches and the stroke of the piston is 16 inches. The steam is cut off at 0·4 of the stroke, the back pressure is 16 lb. per square inch, and the speed of the crank shaft is 100 revolutions per minute. Neglecting clearance and compression, compute the horse-power H and the indicated steam per hour W for the following initial pressures p_1 lb. per square inch, namely, 120, 80, and 40, taking the densities of the steam in lb. per cubic foot at these pressures as 0·268, 0·183, and 0·095 respectively. Show by plotting that the relation between W and H is very approximately of the form $W = mH + k$, and find the best values of m and k.

6. Same as the preceding exercise, except that there is a clearance of 8 per cent. and compression up to 40 lb. per square inch in each case.

CHAPTER XV

INDICATORS AND INDICATOR DIAGRAMS

200. Indicators.—The term *indicator* as used in connection with heat engines, fluid pressure machines, and pumps, has a special and well recognized meaning and is the name given to the instrument which produces a graphic record of the pressure of the fluid in a cylinder for every position of the piston, bucket, or plunger as it reciprocates.

The indicator was invented by Watt and as used by him it consisted of a small cylinder fitted with a piston the under side of which was placed in communication with the cylinder to be "indicated." The upper side of the indicator piston was in communication with the atmosphere. A helical spring on top of the piston had one end attached to the piston and the other to the cover of the indicator cylinder through which passed the piston rod which carried a pencil at its upper end. As so far described the instrument was a pressure gauge and the position of the pencil, at any instant, in relation to its zero position, was a measure of the pressure in the engine cylinder at that instant. The remainder of the instrument consisted of a board which reciprocated in a frame, its motion being that of the engine piston on a reduced scale. This reciprocating board had a sheet of paper attached to it and the pencil on the indicator piston rod traced out on the reciprocating paper the *indicator diagram.*

Apart from small details of construction ordinary modern indicators differ from the Watt indicator in two main particulars.

In the first place the pencil is not now attached directly to the piston rod but to some point in a mechanism of levers which enables the stroke of the piston to be very considerably reduced while still obtaining a convenient height of diagram. This alteration in the design of the Watt indicator became necessary as engine speeds increased. The object of reducing the stroke of the piston is to reduce the inertia effects, which become serious at high speeds, and the diagram is then no longer an accurate pressure record.

The design of the mechanism for making the travel of the pencil greater than that of the piston is one of the principal features which distinguish one modern indicator from another.

The second important change in the design of the Watt indicator was the substitution of a drum for carrying the paper in place of the reciprocating board. This drum has an angular reciprocating motion corresponding to that of the engine piston. This alteration made a more convenient and compact arrangement.

201. Pencil Mechanisms of Modern Indicators.—The principal designs of the mechanism connecting the piston and pencil of modern indicators are shown in Figs. 315 to 320. In each case P is the position of the pencil and the whole pencil mechanism is carried by a piece A which is mounted on the top of the indicator cylinder and is free to swivel about the axis of that cylinder. The object of this is to permit of the pencil point being placed in contact with the paper on the drum when a diagram is to be drawn and to remove the pencil clear of the paper when the diagram is finished. This operation of swinging round the pencil mechanism as a whole is performed by hand. An adjustable stop is generally provided so that when the mechanism is swung round

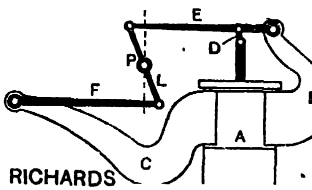


Fig. 315.

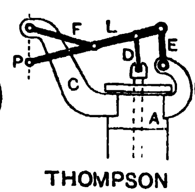


Fig. 316.

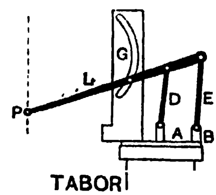


Fig. 317.

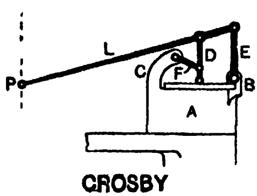


Fig. 318.

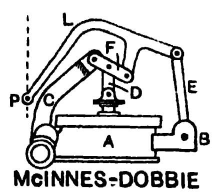


Fig. 319.

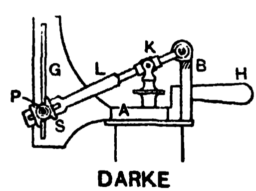


Fig. 320.

to bring the pencil point on to the paper there is just sufficient pressure between the pencil and paper to cause the pencil to mark the paper.

In each design there is an arm or standard B attached to or forming part of the piece A and which forms at its outer end the fulcrum for the radius rod E or, in the case of the Darke design, the fulcrum for the pencil lever L. In all except the Tabor and Darke designs there is a second arm C formed with or attached to A and which forms at its outer end the fulcrum for the radius rod F.

In all except the Darke design the top of the piston rod is connected to the pencil mechanism through the link D.

The pencil is attached in each design to the pencil lever or link L at P.

In the Tabor design the straight line motion of the pencil is ensured by means of the curved slot in the guide plate G. A pin with a roller on it and attached to L works in this slot.

In the Darke design the top of the piston rod is connected by a pin joint to the block K through which the lever L slides as it oscillates. The pencil is attached to the piece S which is guided in the straight slot in the guide plate G. The piece S is carried by the lever L which slides through S as it oscillates.

The chief requisites for a perfect pencil mechanism are:—

(1). The pencil should travel in a straight line parallel to the axis of the indicator piston.

(2). Any small displacement of the pencil in any position should bear a constant ratio to the corresponding displacement of the piston.

(3). The moving parts should be as light as possible consistent with strength and stiffness.

(4). The various joints should work with the minimum of friction and slackness.

Practically all indicators comply with (1) and (2) with sufficient accuracy for ordinary purposes. Of the mechanism shown in Figs. 315 to 320 the Tabor will comply with (1) if the curved slot is correctly made. The Darke mechanism is the only one that is mathematically perfect as regards both (1) and (2). The friction in the neighbourhood of the pencil, however, acts at the maximum leverage in this design. None of the other designs is mathematically perfect as regards (1) or (2).

Most commonly the travel of the pencil is to the travel of the piston as 6 to 1.

202. Indicator Cock.—Before describing in detail some modern indicators reference must be made to a simple but important detail,

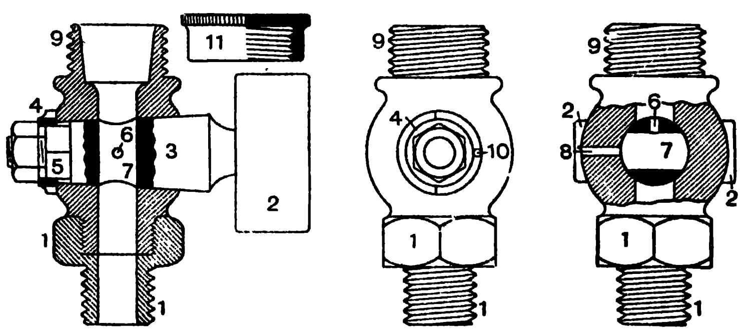


FIG. 321. FIG. 322. FIG. 323.

namely, the *indicator cock* which forms the connection to the engine cylinder.

A good design of indicator cock is shown in Figs. 321, 322, and 323. The part 1 is made of steel and is screwed into a convenient part of the engine cylinder or cylinder cover and communicates with the clearance space. The remainder of the cock is made of gun-metal but the handle 2 of the plug 3 is generally faced with wood or other suitable material which is a bad conductor of heat. The washer 4 fits on a square part 5 on the plug and it therefore turns with the plug. Through one side of the plug there is a small hole 6 leading into the main hole 7 of the plug. Through one side of the body of the cock there is a small hole 8 as shown in Fig. 323.

The indicator is attached to the cock by means of a screwed coupling, which is part of the indicator, and which engages with the screw 9.

There are three positions into which the plug 3 may be turned by hand: (1) Shown in Fig. 323 in which communication with the engine cylinder is cut off and there is free communication between the atmosphere and the indicator through the holes 8, 7, and 6. (2) Shown in Fig. 321 in which there is free communication between the engine cylinder and the indicator but communication with the atmosphere is shut off. (3) The blow-through position in which the plug is turned through 180° from position (1), there being communication between the engine cylinder and the atmosphere but all communication with the indicator is cut off.

For positions (1) and (3) the handle 2 is horizontal while for position (2) it is vertical. The plug is prevented from taking the other position in which the handle is vertical, and in which there would be communication between the engine cylinder and the indicator and also with the atmosphere, by means of the stop pin 10 fixed in the body of the cock and projecting into a gap in the outside edge of the washer 4 and which extends as far as will permit half a turn of the plug.

A cap 11 is screwed on to 9, when the indicator is removed, to keep out dirt.

203. Crosby Indicator.—An instrument which has been highly appreciated for many years is the *Crosby indicator*, one form of which is shown in Fig. 324. For the four blocks used in illustrating this Article the author is indebted to the Crosby Valve and Engineering Co., Ltd., London. This indicator is made in America and has a world-wide reputation.

Referring to Fig. 324, the cylinder 4 in which the piston 8 moves is fixed at the top between the casing 5 and the body A and is thus free to expand or contract. The space between the cylinder and casing serves as a steam jacket. The piston rod 10 is of steel and is made hollow for lightness; at its lower end it is screwed into a socket on the piston and at its upper end it is connected to the pencil lever 16 through the link 14 and the swivel head 12 the lower part 11 of which is screwed into the piston rod. The radius link 15 is jointed at one end to an intermediate point 20 in the link 14 and at the other end to the arm X on the sleeve 3. The fulcrum 18 of the pencil lever is at the upper end of the fulcrum rod 13 which is jointed to the sleeve 3 at 17.

The cover or cap 2 screws into A and prevents the sleeve 3 from rising. In the centre of the cap there is a hole fitted with a hard steel bush which forms a guide for the piston rod. The double helical spring above the piston is connected to the piston at its lower end and to the cap 2 at its upper end. Details of the spring and piston will be further considered presently.

On the body A is an arm 1 which carries the drum 24 and its connections. The central spindle 28 of the drum is secured to the arm 1 by means of the screwed extension and the nut 39. Between the washer 38 and the arm 1 is clamped the piece 33 carrying the pulleys which guide the cord on to the pulley 27 at the base of the drum.

The helical drum spring 31 is attached to a ring at its lower end

which in turn is connected to the drum pulley 27 by the screw 32. At its upper end the drum spring is attached to a cap having a square central hole which fits on to a square part formed on the spindle 28. The torque exerted by the spring on the drum is adjusted by lifting the spring cap clear of the square on the spindle and turning the cap round and dropping it back again over the square on the spindle. The paper is held on the drum by the clips 25. The drum shell 24 is readily detachable from its base.

Taking hold of the knob 22 the sleeve 3 carrying the pencil

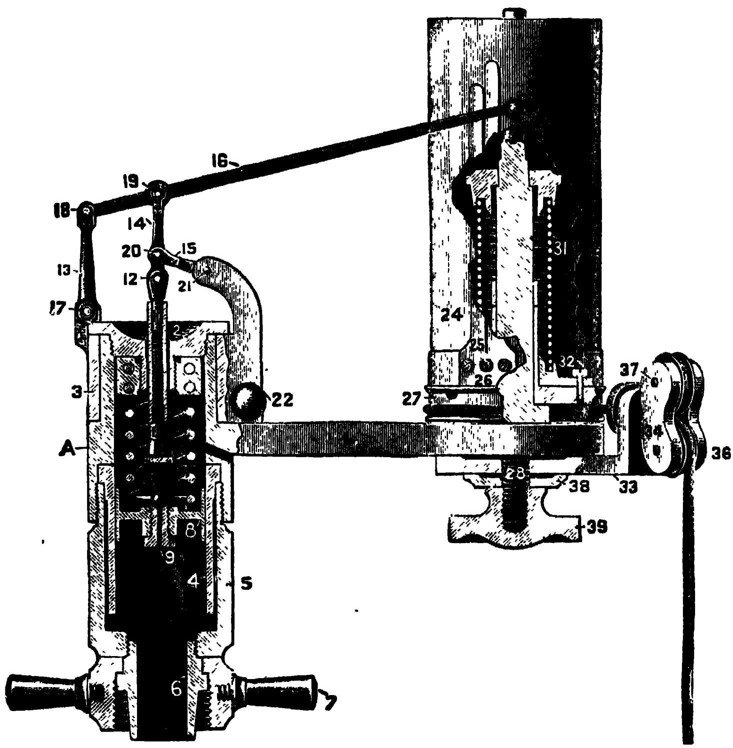


FIG. 324.—Crosby indicator.

mechanism may be turned freely about the axis of the cylinder and the pencil 23 brought into contact with the paper on the drum or removed from it. To prevent undue pressure of the pencil on the paper there is a stop on the arm 1 against which the point of the screw of which 22 is the head comes in contact. The pressure of the pencil on the paper is adjusted by means of the screw just mentioned.

The part 6 fits into the top of the indicator cock, not shown in Fig. 324, and is secured to the latter by the coupling nut 7.

Before the indicator cock is turned on, the piston 8 is under the pressure of the atmosphere on top and bottom and the line traced on

the paper for this condition of the piston is the *atmospheric line.* The position of the atmospheric line on the paper is adjusted by means of the screw 11—on the lower part of 12. The cap 2 is unscrewed and the sleeve 3 with its connections is lifted from the cylinder, the piston with the parts connected to it is then rotated so as to bring the joint 12 nearer to or further from the top of the piston rod as may be required.

The piston spring is shown separately in Fig. 325. It is made from a straight piece of steel wire in the middle of which is fixed a steel bead. The wire is then wound from the middle to form two helices the ends of which are passed through holes drilled helically in four radial wings on the nut which screws on to the cap 2 in Fig. 324. Adjustment of the stiffness of the spring is made by screwing it more or less into the wings of the nut before it is finally fixed thereto.

The attachment of the foot of the spring to the piston is shown separately in Fig. 326 where A is the piston and B the piston rod. The bead on the spring is held between the lower end of the piston rod and the upper end of the screw which screws into the lower socket on the

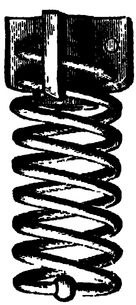


Fig. 325

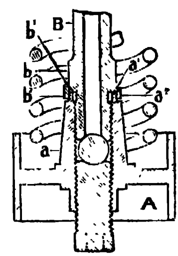


Fig. 326.

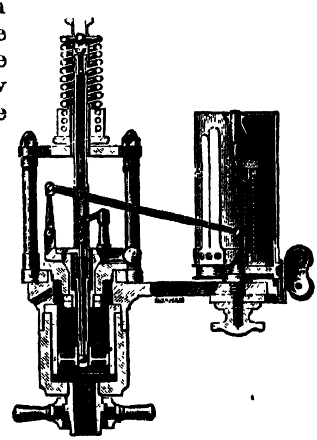


Fig. 327.

piston. To permit of the bead on the spring entering the upper socket a on the piston that socket is slotted to clear the wire of the spring on each side of the bead. To prevent the slotted socket from opening when the piston rod is screwed into it there is a socket a' b' formed in the shoulder of the enlargement b of the piston rod to receive the spigot a^2 b^2 formed on the top of a. It is most important that the screw below the bead should not be screwed up against the bead until the piston rod has been firmly screwed home and the screw below should then be screwed up lightly against the bead.

The Crosby design of piston spring and its connection to the piston is ingenious and admirable; it ensures that the thrust of the spring on the piston is central and that there is no tendency to tilt the piston and thus cause friction between the piston and cylinder.

Another design of Crosby indicator is shown in Fig. 327. This has an *external spring* which is in tension instead of compression. The great advantage of the external spring type of indicator is that the spring is not affected by the changes of temperature which occur in

the cylinder of the indicator, especially when the indicator is used on an internal combustion engine.

A radical departure in the design of the piston is also shown in Fig. 327. The piston is in the form of the central zone of a sphere and the contact between it and the cylinder is therefore line contact.

For steam engine work the piston of an indicator has, most commonly, an area of half a square inch, but for indicating internal combustion engines, in which the maximum pressure is much higher than in steam engines, the area of the piston is one-quarter of a square inch.

204. Dobbie McInnes Indicator.[1]—Another indicator with a high reputation is the *Dobbie McInnes indicator.* This instrument is made

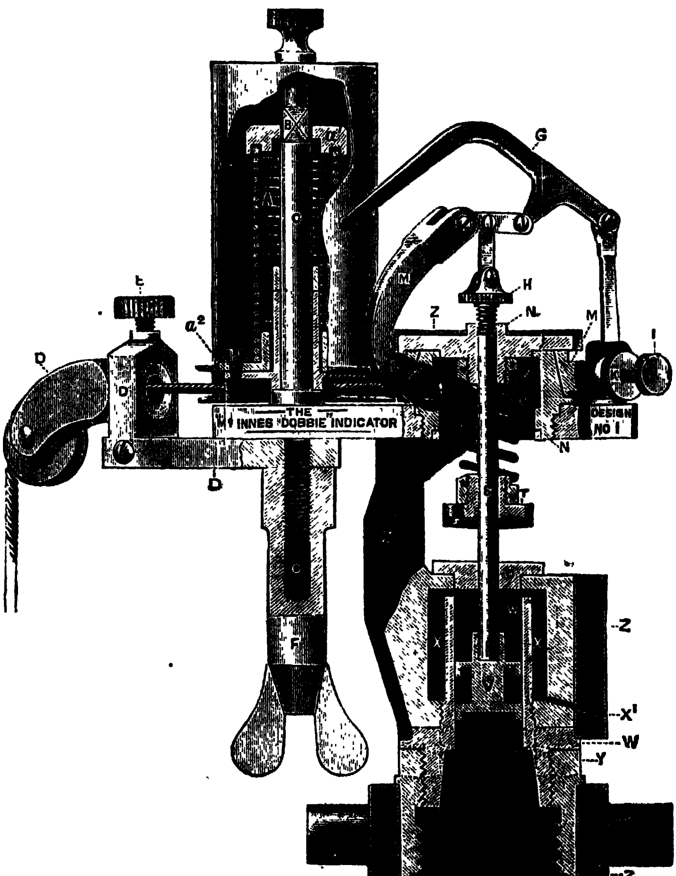


FIG. 328.—Dobbie McInnes Indicator.

by Messrs. Dobbie McInnes, Ltd., of Glasgow, to whom the author is indebted for the illustration, Fig. 328, which shows the external spring type of their indicator.

[1] Formerly known as the "McInnes Dobbie" indicator.

The helical drum spring A has brass ends a^1 and a^2 of which a^1 fits on the square B on the drum spindle C, while a^2 reciprocates with the drum. To increase or diminish the torque of this spring on the drum the top end a^1 is lifted off the square B and turned to left or right and then replaced. D is the frame, bracket and stud, carrying the cord guide pulley whose position may be adjusted to take the cord off at any angle by means of the set screw E and clamping fly nut F. The pencil mechanism G multiplies the piston travel six times at the pencil point. H is a milled nut connecting the piston rod S to the pencil mechanism. The pressure of the pencil on the paper is adjusted by means of the screw pin I which is locked in position by the binding nut K. The vulcanite covered handle L is used to manipulate the swinging arm or bracket M carrying the pencil mechanism. N is the upper cylinder cover or cap, around which the bracket M is fitted and swings. The cap N is flanged above the centre of the bracket M, holding this so that when the cap is unscrewed the cap and bracket are still connected. The lower cylinder cap O prevents steam escaping upwards and affecting the pressure spring U or moistening and damaging the diagram card. Any steam passing the piston V is discharged to the atmosphere through the escape hole X^1 at the bottom of the cylinder.

When the top cover N is unscrewed, the piston cap O, piston V, swinging bracket M, and pencil mechanism are all released and pull up together. The pressure spring U has its upper end screwed to the top cover N and its lower end screwed to the spring seat T which is fixed to the piston rod. The piston V is made of steel and has a recess for lubricant and the accommodation of grit. The nut Y serves to connect the indicator to the indicator cock. The vulcanite sheathing Z prevents burning of the fingers of the operator when the indicator is under steam.

205. Optical Indicators.—For engines running at high speeds, say over 300 revolutions per minute, the ordinary indicator with its linkwork pencil motion is unsuitable on account of the inertia of its reciprocating parts. For this reason, in practice, the indicated horsepower of high-speed engines is, generally, not determined and the performance is based on the brake or shaft horse-power.

For scientific work on high-speed engines it is, however, important to know what is going on in the cylinder, as far as can be learned from an indicator diagram, and for this purpose *optical indicators* have been devised.

In an optical indicator a very stiff spring having a small deflection for the range of pressure in the cylinder is used, and this small movement is communicated to a tilting mirror upon which a pencil of light is projected and then reflected on to the photographic plate of a camera. At the same time the mirror is given an oscillating motion about an axis at right angles to the axis of tilt to correspond to the motion of the engine piston.

In some optical indicators a thin metal diaphragm is used instead of a piston and spring, the diaphragm being clamped round its circumference.

One well-known optical indicator is that invented by the late

Professor Bertram Hopkinson, of Cambridge. This instrument, which is made by Messrs. Dobbie McInnes, Ltd., of Glasgow, is shown in Fig. 329. A block A is screwed into the ordinary indicator hole of the engine. The frame B fits over the block, sufficient clearance being left to provide for unequal expansion. The frame is held up by a spring into engagement with the lower face of the nut C, screwed to the top of A, a ball-race being interposed so as to admit of easy rotation of the frame about the axis of A.

The spring D is a piece of steel strip resting in grooves at the ends of the frame B, and held by the screws E. The spring is slightly bowed before insertion in the frame, so that when the screws E are

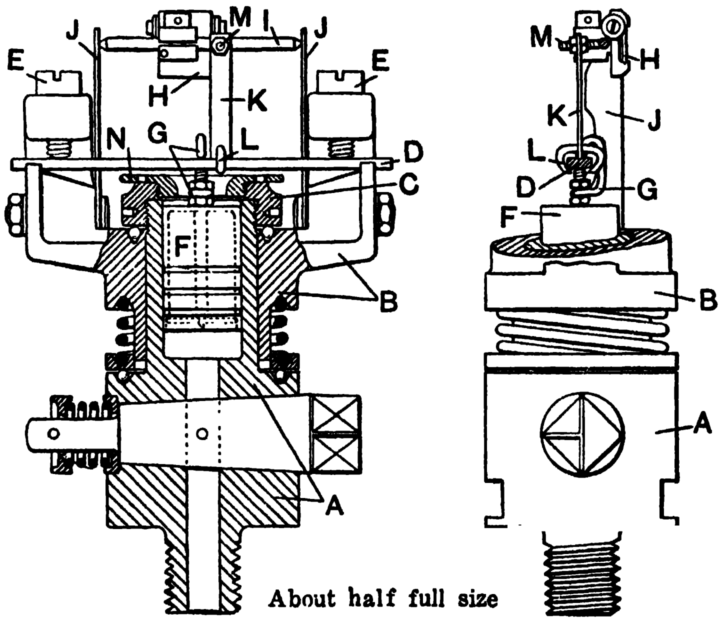


FIG. 329.—Hopkinson optical indicator.

screwed home, the spring is held straight with slight pressure on the four points of support.

The piston F slides in a bore in the block A. At the top it is provided with a hook G, the opening of which is slightly larger than the thickness of the spring. The piston is thus free to move laterally, and no binding action is possible between it and the sides of the bore, such as would occur if the piston were rigidly attached to the spring.

Three pistons are supplied, the areas being in the ratio of 1, 2, and 4. There are two springs which are ground so that their stiffnesses are in the ratio of 1 to 5. A wide range of sensibility is thus obtained. The smaller pistons fit inside liners which are inserted in the bore of the block A.

The mirror H is clamped to a steel spindle I, the ends of which are pivoted in small holes in the vertical spring cheeks J. The motion

of the spring D is communicated to the spindle and mirror by means of the piece of vertical spring K. The lower end of this spring is held firmly on the face of the main spring D by means of the jaws L, the upper end is firmly clamped to the arm M, which projects at right angles from the mirror spindle.

The spring K, while sufficiently rigid to transmit the motion of the main spring to the end of the arm M without buckling, is flexible enough to allow for the angular motion of that arm. The mirror is thus turned about the axis of the spindle by an amount which is proportional to the displacement of the main spring D, and therefore to the pressure under the piston.

The upward motion of the piston is limited, and excessive bending of the spring D prevented, by the cap N.

In order to give the other motion to the mirror, the frame B is positively connected by linkage to a reciprocating part of the engine, and is thus caused to oscillate as a whole about the axis of the block A. The motion thus given to the frame B must be in phase with and proportional to the piston motion.

The camera supplied is readily fixed to or removed from the indicator.

206. Reducing Gears.—The stroke of the engine piston being generally much greater than the length of the required indicator diagram some form of reducing gear is necessary to connect the indicator drum with the engine crosshead or some other part which has the same motion as the piston. The designs of reducing gears for this purpose are exceedingly numerous, a good many being correct while the others are only approximately so. But many of those which are not correct are sufficiently exact for practical purposes, and if they are simple in construction great refinement of accuracy may generally be dispensed with.

A reducing gear is said to be correct when the linear motion of the paper on the indicator drum is an exact copy, on a reduced scale, of the motion of the engine piston, that is to say, when the paper has made 1-nth of its travel the piston must have made 1-nth of its stroke, where n is any number.

The most common form of reducing gear is the oscillating lever type, four examples of which are shown in Figs. 330 to 333.

Referring to Fig. 330, a lever BC is hung from a fixed pin C. This pin C is attached to a bracket or standard fastened to a convenient part of the engine frame. A pin A attached to the engine crosshead projects therefrom into a slot in the lever at B. Aa is the line of stroke of the pin A, a is its middle position, and Ca is perpendicular to Aa. As the pin A reciprocates with the engine crosshead the lever BC swings about the pin C. One end of the indicator cord is attached to the pulley at the base of the drum and is coiled round that pulley while the other end is attached to the lever BC at E. Cca is the centre line of the lever in its middle position. For the sake of clearness the upper part of Fig. 330 is enlarged as a simple line diagram at (m). XE parallel to aA is the displacement of E in that direction from its middle position. $\frac{XE}{aA} = \frac{CX}{Ca}$, but CX is not a fixed length,

therefore the ratio of XE to aA is variable and depends on the position of the lever. The position of E when the lever is in the middle of its swing is e. Draw eY parallel to aA, then $\frac{eY}{aA} = \frac{Ce}{Ca}$ which is constant. Hence eY instead of XE should be the displacement of E parallel to aA for a correct reducing gear.

If the indicator cord proceeds from E in the direction parallel to aA, then XE is the length of cord wound on to the drum pulley for a displacement aA of the engine piston, and this is less than it should be for a correct gear by the difference between eY and XE. If the cord from E is not parallel to aA it is easy to show that the error will be increased.

Now let a sector CRS, Fig. 331, be fixed on the lever BC, the centre of the pin C being the centre of the circle of which RS is an arc, and let the cord be attached to this sector so that as the lever swings the cord is always tangential to the arc RS. Then for the

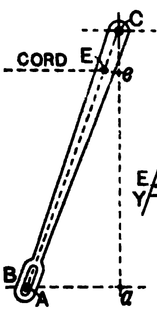


FIG. 330.

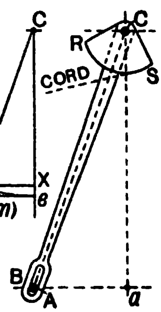


FIG. 331.

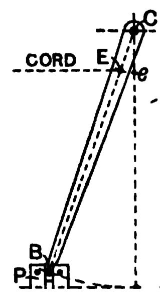


Fig. 332.

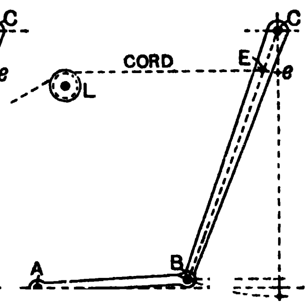


FIG. 333.

displacement shown the length of cord unwound from the sector is the length of the arc eE, which is greater than XE but less than eY. The addition of the sector CRS, called a *brumbo pulley*, has therefore improved the accuracy of the gear. Also, the cord may be taken off at any inclination, so long as it is tangential to RS, without increasing the inaccuracy.

By attaching a slotted plate P, Fig. 332, to the crosshead, the slot being at right angles to the line of stroke, and by fixing in the lever the pin B which works in the slot, the displacement of E parallel to the line of stroke will bear the constant ratio CE : CB to the crosshead displacement, and if the cord from E is parallel to the line of stroke the reducing gear becomes almost perfect. The only error is now due to the small displacement of E at right angles to the line of stroke which prevents the cord from E remaining perfectly parallel to the line of stroke. In general this error is quite negligible.

Another oscillating lever gear which is nearly perfect is that shown in Fig. 333. The pin A is fixed to the crosshead and projects into the link AB. This link is jointed to the swinging lever BC at B. But for the small oscillations of the link AB about A the displacement of E parallel to the line of stroke would bear the constant ratio CE : CB

to the crosshead displacement. This gear has been much used on gas engines.

The cord in Figs. 330, 332, and 333, is guided by a pulley L (Fig. 333), so that the part of the cord between E and the pulley shall be practically parallel to the line of stroke.

The gear shown in Fig. 332 may be made perfect by altering it to the form shown in Fig. 334. A slider D is guided parallel to the line of stroke. A pin E attached to the slider projects into a slot T in the swinging lever. The motion of the slider is an exact copy of the motion of the crosshead on a reduced scale. The cord is attached to the slider and leaves it in a direction parallel to the line of stroke.

In another type of reducing gear two pulleys of different diameters

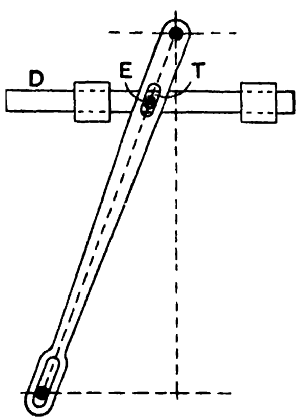


FIG. 334.

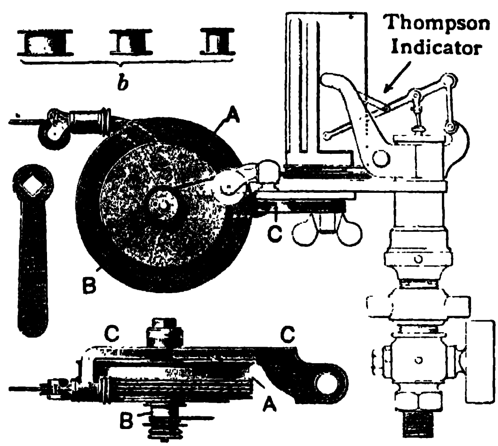


FIG. 335.

and connected together are mounted on the same axle. A cord connects the larger pulley with the engine crosshead and another cord connects the smaller pulley with the drum pulley of the indicator. The reduction of motion is the ratio of the diameter of the smaller pulley to that of the larger. This type of gear gives a constant reduction of motion throughout the stroke.

An example of the pulley type of reduction gear is shown in Fig. 335 attached to a Thompson indicator. The larger pulley A and the smaller pulley B rotate together on an axle carried by the frame C. The pulley B may readily be detached and replaced by another one of a set having different diameters so as to adapt the reduction of motion to the stroke of the engine piston. Three extra pulleys are shown at (*b*). The pulleys are made of thin hard-rolled aluminium to reduce the weight to a minimum and make the gear suitable for high speeds.

207. Mean Effective Pressure from Indicator Diagram.—One of the factors required in determining the work done on the piston of an engine per revolution of the crank shaft, as a step to finding the

indicated horse-power, is the mean effective pressure on the piston. The M.E.P. is the mean height of the indicator diagram measured on the pressure scale of the diagram.

There are various ways of finding the mean height of a diagram. The best way, when a *planimeter* is available, is to measure the *area* of the diagram with that instrument and then dividing the area found by the extreme length of the diagram, measured parallel to the atmospheric line, the result is the mean height, and this multiplied by the number of the pressure scale (units of pressure per unit of length) gives the required M.E.P. For example: if the area of the diagram as found by the planimeter is 1·68 square inches, and if the length of the diagram is 3 inches, then the mean height of the diagram is $1·68 \div 3 = 0·56$ inch, and if the pressure scale is 100 lb. per square inch per inch, then the M.E.P. is $0·56 \times 100 = 56$ lb. per square inch.

The *mid-ordinate method* is the one generally used when a planimeter is not available. This method is illustrated by Fig. 336. The length of the diagram is divided into a number of equal parts, generally 10, and these parts are bisected by ordinates perpendicular to the atmospheric line AL. Applying the pressure scale the lengths of the ordinates lying within the diagram are measured and noted on the diagram as shown; adding these up and dividing by the number of ordinates, in this case 10, the M.E.P. is found.

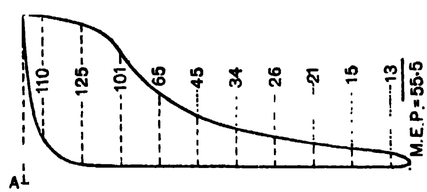


Fig. 336.

Instead of using the pressure scale directly on the separate ordinates, the lengths of these may be marked off in succession and continuously on the straight edge of a strip of paper. The sum of the ordinates is then measured, say, in inches, and the result multiplied by the number of the pressure scale and divided by the number of ordinates gives the M.E.P.

Theoretically the M.E.P. found by means of ordinates will be more accurate the larger the number of ordinates used, but in practice the number need not exceed 20.

If the length of the diagram be divided into as many as 20 equal parts and ordinates be drawn through the points of division and also through the end points there will be 21 ordinates in all and instead of drawing and measuring the mid-ordinates the mean ordinate may be found by first finding the mean of all the ordinates except the last, then the mean of all except the first, and finally the mean of these two means which is the mean required.

208. M.E.P. Referred to L.P. Cylinder.—If the total work done in the two or more cylinders of a compound or a multi-stage expansion engine be assumed to be done in the low-pressure cylinder alone, then the necessary M.E.P. in that cylinder is called the *mean effective pressure referred to the low-pressure cylinder.*

Let A denote the mean area of the L.P. piston and A_1, A_2, etc. the mean areas of the other pistons. Also, let L, L_1, L_2, etc. denote the

respective strokes and p, p_1, p_2, etc. the respective mean effective pressures. Then if p_m = M.E.P. referred to L.P. cylinder,

$$p_m AL = pAL + p_1A_1L_1 + p_2A_2L_2 + \text{etc.}$$

209. Mean Indicator Diagram.—When an important trial of an engine is made numerous indicator diagrams are taken and it is important to have for each cylinder a diagram which is the mean of all those taken from the same cylinder.

An obvious, but somewhat laborious, way of obtaining a mean diagram is to draw similar sets of ordinates on the diagrams to be averaged and then find the means of corresponding ordinates as the ordinates of the mean diagram. All the ordinates are measured from the atmospheric line to which they are perpendicular.

An approximate mean diagram may be more easily found by measuring the areas of all the diagrams to be averaged with a planimeter and then, after finding the mean of all the areas, select that diagram whose area is nearest to the mean area, and take that as the mean diagram.

A method, which seems to be an admirable one, is to make a composite photograph of the diagrams. This method is fully described by Mr. George E. Scholes in *Engineering*, Jan. 10, 1913, and is carried out by photographing the diagrams in succession on the same plate, giving each 1-nth of the correct exposure under the lighting conditions, where n is the number of diagrams to be photographed,

210. Modifications of Theoretical Indicator Diagram in Practice.—The features of the theoretical indicator diagram for a steam engine, which is used for comparison with the actual diagram, are exhibited at (O) in Fig. 337. The steam is admitted at F and the pressure in the cylinder rises at once to the full pressure as shown by the vertical *admission line* FA. Admission continues at constant pressure as shown by the horizontal *steam line* AB until the *point of cut off* B is reached. Cut off takes place instantaneously. Expansion of the steam then takes place and continues until the piston reaches the end of its forward stroke. This operation is represented by the *expansion curve* BC. For the purpose in view it is usual to assume that the curve BC is a rectangular hyperbola whose asymptotes are OX and OY, the axes of volume and pressure. This means that PV = constant. This is, however, not strictly true even for an ideal diagram. The exhaust begins at C, the *point of release*, and the pressure falls at once to the back pressure as shown by the vertical *exhaust line* CD. Exhaust continues at constant pressure, as shown by the horizontal *back pressure line* DE. At E the exhaust is stopped and *compression begins* and continues to the end of the return stroke as shown by the *compression curve* EF. The compression curve is assumed to be another rectangular hyperbola having the same asymptotes as BC.

The principal modifications of the above features which appear on actual indicator diagrams are shown in the sketches (1) to (8) in Fig. 337.

(1). The steam line AB usually slopes downwards as Ab. This is due to the throttling action of the steam port. Instead of being straight, as shown, Ab may be more or less curved.

(2). The cut off is more or less gradual, beginning at b and finishing at b'. This causes a gradual throttling action which produces the curve bb'. The virtual point of cut off is B where the steam and expansion lines meet when continued.

(3). The actual expansion curve Bc may fall below BC either as the result of leakage past the piston or of condensation in the cylinder. If the actual curve Bc' is above BC this may indicate steam valve leakage or it may be due to the action of the steam jacket if there is one.

(4). Release takes place at c before the end of the forward stroke is reached and it takes place more or less gradually. The pressure

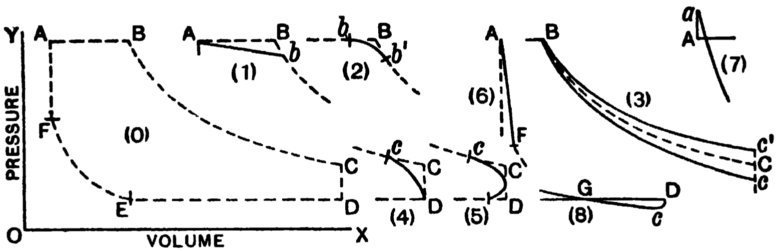


FIG. 337.

may fall to the back pressure before the end of the stroke; at (4) the exhaust pressure is shown reaching the back pressure at D, the end of the stroke.

(5). The exhaust pressure does not reach the back pressure until part of the return stroke of the piston has been performed.

(6). If the admission valve opens before the beginning of the forward stroke the admission line FA will slope as shown.

(7). This shows the compression curve rising above the initial steam pressure before the admission valve opens. This causes a loop in the diagram.

(8). A loop may also be formed by a very early cut off; the steam then expands to below the back pressure before release. The expansion line is shown crossing the back pressure line at G.

The area of a loop is to be taken as negative when determining the area of the diagram.

211. Effects of Speed on Indicator Diagrams.—With the same valve setting an indicator diagram will have different shapes at different engine speeds. The higher the speed the greater is the throttling action on the steam in passing through the ports and passages. There is also less interchange of heat between the cylinder and the steam during expansion and compression at the higher speeds.

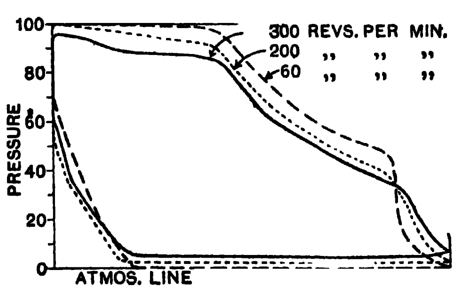


FIG. 338.

The effect of engine speed on the indicator diagram is well shown in Fig. 338, copied from one given by the late Dr. J. T. Nicolson in the *Proceedings of the Institution of Mechanical Engineers*, 1905, p. 335. Three superposed diagrams are shown for crank shaft speeds of 60, 200, and 300 revolutions per minute, the valve setting being the same for all.

212. Indicated Steam and Missing Quantity.—The determination of the steam consumption of an engine from an ideal indicator diagram has been explained in Art. 197, p. 275. The subject will now be further considered with reference to an actual indicator diagram and the actual performance of an engine.

A mean indicator diagram from the low-pressure cylinder of a compound engine is given in Fig. 339. The stroke volume KL of the cylinder is 43·3 cubic feet and the clearance volume OK is 2·2 cubic feet. A point A is selected on the expansion curve. At this point the absolute pressure is 16 lb. per square inch and the volume OM is 22·77 cubic feet. Assuming that the steam in the cylinder to the left of A is dry and saturated its weight is $\frac{22\cdot77}{24\cdot74} = 0\cdot920$ lb., where 24·74 is the volume, in cubic feet, of 1 lb. of dry saturated steam at the pressure AM.

A point B is next selected on the compression curve, near to where compression begins. At this point the absolute pressure BN is 4 lb. per square inch and the volume ON is 4·88 cubic feet. Assuming that the steam in the cylinder to the left of B is dry and saturated, its weight is $\frac{4\cdot88}{90\cdot5} = 0\cdot054$ lb., where 90·5 is the volume, in cubic feet, of 1 lb. of dry saturated steam at the pressure BN. This is the weight of the *cushion steam* or the weight of the steam which gets compressed into the clearance space.

The difference between the weight of the steam at A as found above and the cushion steam is called the *indicated steam* per stroke for the point A.

During a trial of the engine the actual steam consumption was 13,680 lb. per hour and the average speed was 100·1 revolutions per minute. Hence the weight of steam supplied on each side of the piston per revolution was $\frac{13{,}680}{60 \times 100\cdot1 \times 2} = 1\cdot139$ lb. This is called the *cylinder feed*. Adding this amount to the cushion steam the actual weight of steam in the cylinder to the left of A is $0\cdot054 + 1\cdot139 = 1\cdot193$ lb. But from the indicator diagram it has been computed that the amount of steam in the cylinder to the left of A is only 0·920 lb. The difference between these two amounts, $1\cdot193 - 0\cdot920 = 0\cdot273$ lb., is known as the *missing quantity* of steam per stroke at the point A. Neglecting leakage the missing quantity is accounted for by condensation. Hence the steam at A is not dry and its dryness fraction is $\frac{0\cdot920}{1\cdot193} = 0\cdot771$.

It will be noticed that the actual weight of steam at A, 1·193 lb., was obtained by adding the cushion steam to the cylinder feed, and in

computing the cushion steam it was assumed to be dry and saturated which is probably not strictly true. But at B the temperature of the cylinder cannot differ much from that of the steam, and the water in the steam during exhaust would probably pass to the condenser. In any case the weight of the cushion steam is generally small compared with the cylinder feed and therefore any error in regard to the dryness of the cushion steam will have little influence on the dryness fraction of the steam at A as calculated above.

213. Saturation Curve on Indicator Diagram.—Still referring to Fig. 339, it has been shown that for the point A there is 1·193 lb. of steam and water in the cylinder. Neglecting the volume of the water, the volume of the steam is OM which represents 22·77 cubic feet, but if all the water and steam present were dry saturated steam the volume would be $24{\cdot}74 \times 1{\cdot}193 = 29{\cdot}51$ cubic feet. Now on the horizontal line through A make $RS = 29{\cdot}51$ on the volume scale of the diagram, then S is a point on the saturation curve. The distance AS is the volume of the missing quantity and the ratio of RA to RS is the dryness fraction of the steam in the cylinder for the point A.

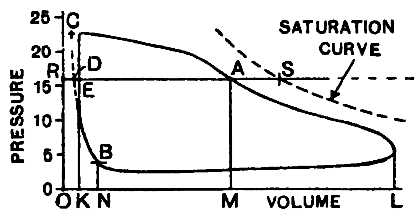


FIG. 339.

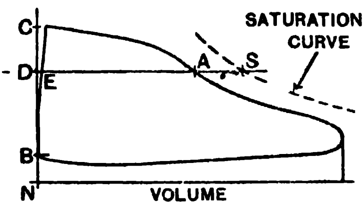


FIG. 340.

Any number of points on the saturation curve may be found by taking from the steam table the volumes of 1 lb. of dry saturated steam at selected pressures and multiplying these by 1·193 to obtain the volumes for the saturation curve at the selected pressures.

If all the steam in the cylinder were dry and saturated during expansion the expansion curve would lie on the saturation curve.

214. Work Diagram Excluding Cushion Steam.—A diagram representing the action of the cylinder feed by itself may be constructed as follows. Referring to Fig. 339 let the compression curve be continued up to the inital pressure. In doing this it will be sufficiently accurate to assume that the curve follows the law $pv =$ constant. Let this curve BDC cut any horizontal such as RAS at D. Then DA is the volume of the cylinder feed at the pressure AM exclusive of the missing quantity and DS is the volume including that quantity. Now let the points E, A and S, where the horizontal RS cuts the indicator diagram and the saturation curve, be moved to the left by the amount RD, and let this construction be repeated at a sufficient number of different levels. Joining up the new points found another diagram and another saturation curve are obtained which for the sake of clearness are shown separately in Fig. 340.

The new diagram will have exactly the same area as the original one, the volume of the missing quantity will be the same at the same

pressure, but the saturation curve will now be the saturation curve for the actual cylinder feed.

215. Combination of Indicator Diagrams of Compound Engines.—The indicator diagrams as taken from the cylinders of compound engines have different pressure scales, and since the lengths of the diagrams are generally about the same, the volume scales are different owing to the volume of the low-pressure cylinder being much greater than that of the high-pressure cylinder. For some purposes it is convenient to redraw the diagrams using the same pressure scale and the same volume scale for both, and having the same pressure and volume axes. The procedure is simple and is illustrated by the example in Fig. 341, which is taken from one of Mr. Michael Longridge's reports to the British Engine, Boiler, and Electrical Insurance Co., Manchester.

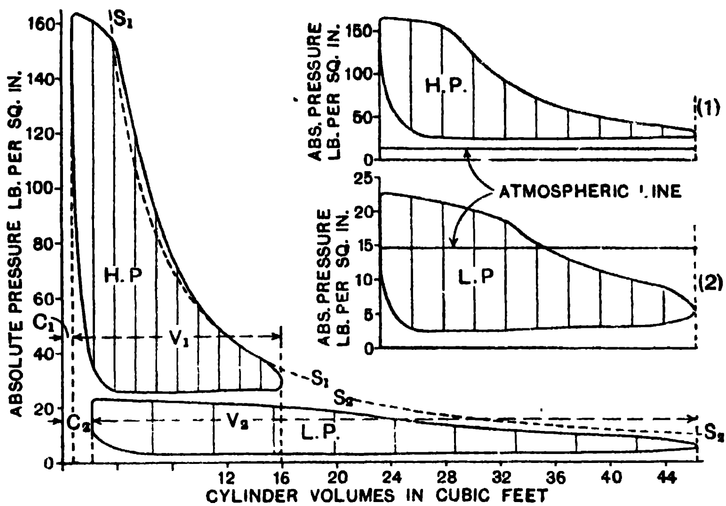


FIG. 341.

The actual indicator diagrams are shown at (1) and (2), reduced in size. The stroke volume of the H.P. cylinder is 14·94 cubic feet, and the clearance volume is 0·75 cubic foot. The stroke volume of the L.P. cylinder is 43·3 cubic feet, and the clearance volume is 2·22 cubic feet.

From ten to twenty equally spaced ordinates are drawn on each indicator diagram. Axes of pressure and volume for the combined diagram are drawn and scales for pressure and volume selected. C_1 and C_2, the clearance volumes of the H.P. and L.P. cylinders respectively, are marked off and to these are added the stroke volumes V_1 and V_2. Two sets of equally spaced ordinates are next drawn in the spaces V_1 and V_2, the number of ordinates in each set being the same as drawn on the actual indicator diagrams. The pressures at the respective ordinates, measured from the line of zero pressure, are read off from the indicator diagrams and measured off on the new diagram to the new scale. The points thus found are then joined up by fair

curves. The saturation curves S_1S_1 and S_2S_2 are generally added as shown.

It will be observed that the saturation curves if continued would not form one continuous curve. This is due to the fact that, although the cylinder feed may be the same in both cylinders, the amount of steam in the H.P. cylinder for which the curve S_1S_1 is drawn is not the same as that in the L.P. cylinder for which the curve S_2S_2 is drawn, the amount of cushion steam in the H.P. cylinder being greater than that in the L.P. cylinder. The cylinder feed to the L.P. cylinder may also be less than that to the H.P. cylinder due to condensed steam in the intermediate receiver being drained off before reaching the L.P. cylinder.

It will be seen that the greater part of the expansion curve for the H.P. cylinder lies outside the saturation curve; this is due to the fact that the steam entering that cylinder is superheated 137° F. (76·1° C.).

It is instructive to draw the combined diagram after eliminating the cushion steam from each indicator diagram in the manner described in Art. 214. In that case if the cylinder feed is the same to both cylinders, which is the case in the example illustrated by Fig. 341, the saturation curves will be continuous.

216. Construction of Temperature-Entropy Diagram from Indicator Diagram.—The first point to be clearly kept in mind in regard to the construction of a temperature-entropy ($T\phi$) diagram on a $T\phi$ chart is that the chart is constructed for 1 lb. weight of steam, or water, or steam and water.

The second point is that it is convenient to assume that the supply of steam, or water, or steam and water used in the engine is used over and over again, and that the operations of heating and evaporating the water, and the subsequent condensation of the steam, take place in the engine cylinder. Even when the engine is non-condensing, or when the steam is exhausted into another cylinder to do further work, it is convenient to assume that the steam which leaves the cylinder during exhaust is condensed in the cylinder, because it is water which is fed into the boiler, and whether this water is the steam from the engine condensed or a fresh supply will not affect the indicator (PV) diagram or the $T\phi$ diagram. Also, during admission, before the whole steam supply for the cycle has entered the cylinder, it will not affect these diagrams if it is assumed that the remainder of the supply is in the form of water in the cylinder.

Stated briefly, any of the steam supply not in the cylinder at any point of the piston stroke is assumed to be water and since the volume of that water is negligible compared with the volume of the steam the measurement of the volumes on the PV diagram is not affected by the assumption that the above mentioned water is in the cylinder with the steam.

The first step to be taken before the $T\phi$ diagram can be proceeded with is to find the weight W of the working steam and water in use in the cylinder during a cycle on one side of the piston. This step has been fully explained in Art. 212. W is made up of the cushion steam and the cylinder feed.

If at any point A of the stroke where the pressure is p the PV

diagram shows that the volume of the steam in the cylinder is V cubic feet, this is for W lb. and the volume for 1 lb. is $v_1 = V/W$. The factor $1/W$ is called the *volume factor.* If v is the specific volume of the steam at pressure p then v_1/v is its dryness fraction. If t is the temperature corresponding to the pressure p, the point on the Tϕ diagram corresponding the point A on the PV diagram can now be fixed.

The procedure in constructing the Tϕ diagram corresponding to a given PV diagram will now be illustrated by an example. The indicator diagram chosen is given in Fig. 342 and is the same as that made use of in Art. 212.

The starting point 1 is taken on the expansion curve and is the same as the point A in Fig. 339, p. 298. The actual weight of steam and water in the cylinder was found in Art. 212 to be 1·193 lb. The volume factor is therefore 1/1·193 and the volumes taken from the

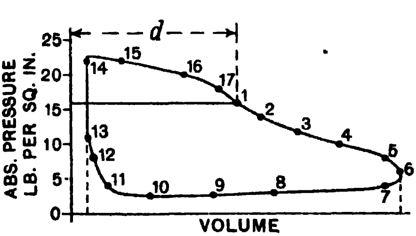


FIG. 342.

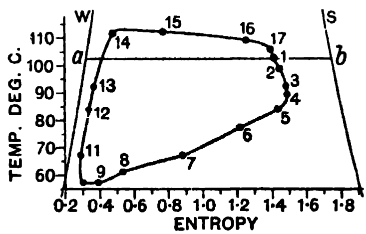


FIG. 343.

indicator diagram must be multiplied by this factor. This means that in the example being considered the Tϕ diagram is to be constructed for an engine cylinder smaller than the actual one in the ratio of 1 to 1·193 by volume.

The volumes from the indicator diagram are best found as follows. For the point being considered measure the distance d, which includes the clearance, say in inches. The stroke length of the diagram, before reduction to Fig. 342, is 3·22 inches and this represents 43·3 cubic feet. The actual volume of the steam in the cylinder for the point considered is $\frac{43 \cdot 3d}{3 \cdot 22}$ and this is for 1·193 lb. of steam. The volume of 1 lb. of steam for this point is $\frac{43 \cdot 3d}{3 \cdot 22 \times 1 \cdot 193} = 11 \cdot 272d = cd = v_1$. For the point 1, $d = 1 \cdot 69$ inches and $v_1 = 11 \cdot 272 \times 1 \cdot 69 = 19 \cdot 05$ cubic feet. The pressure at this point is 16 lb. per square inch absolute and the temperature t is 102·4° C., and the specific volume v is 24·74 cubic feet. Therefore the dryness fraction x is $\frac{19 \cdot 05}{24 \cdot 74} = 0 \cdot 770$.

The point 1 on the Tϕ diagram, Fig. 343, is now found as follows. Draw ab at the level 102·4° C., cutting the water line at a and the saturated steam line at b. Measure ab on any scale and make $a1 = 0 \cdot 770ab$ on the same scale.

If the Tϕ chart has a sufficient number of constant volume lines or

a sufficient number of quality lines on it the point 1 may be spotted without direct measurement of $a1$.

A sufficient number of points on the PV diagram have to be dealt with in the same way as for the point 1 and the points found on the $T\phi$ chart are then joined up.

It will be found convenient to tabulate the various steps for the selected points as shown in the following table, which gives the results for four points only.

Point.	p	t° C.	v	d	$r_1 = cd$	$x = r_1/v$
1	16	102·4	24·74	1·69	19·05	0·770
7	4	67·2	90·5	3·22	36·30	0·401
12	8	83·8	47·28	0·20	2·25	0·048
15	22	111·7	18·37	0·50	5·64	0·307

217. Diagram Factor.—The ratio of the M.E.P. obtained from an actual indicator diagram to the M.E.P. from the corresponding theoretical diagram is called the *diagram factor*. In the theoretical diagram referred to, clearance and compression are neglected and the expansion is assumed to be hyperbolic. The initial pressure is generally taken as the boiler pressure but it is better to take it as the pressure at the engine stop valve. The back pressure is that expected in the actual engine. The expansion ratio taken is the *nominal expansion ratio* which is the ratio of the stroke volume to the volume at cut off, neglecting clearance, in the case of a simple engine, and the ratio of the stroke volume of the low pressure cylinder to the volume at cut off in the high pressure cylinder, neglecting clearance, in the case of a compound or multi-stage expansion engine.

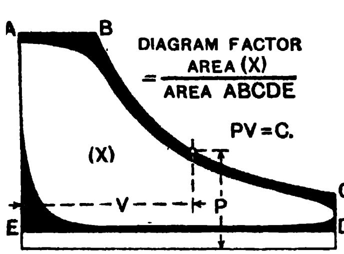


FIG. 344.

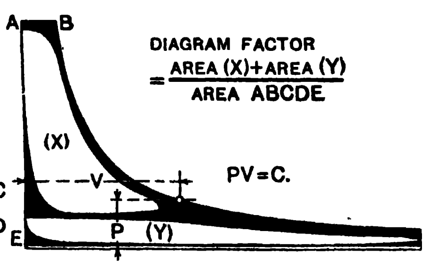


FIG. 345.

Fig. 344 shows the theoretical diagram ABCDE and the actual diagram (X) for a simple engine, and Fig. 345 shows the theoretical diagram ABCDE and the actual diagrams (X) and (Y) for a compound engine.

In compound and multi-stage expansion engines the M.E.P. mentioned above is the M.E.P referred to the low-pressure cylinder.

Commonly the diagram factor is from 0·7 to 0·9 in simple engines, from 0·6 to 0·8 in compound engines, and from 0·6 to 0·7 in triple-expansion engines. It is higher the greater the ratio of expansion.

It is also higher with jacketed than with unjacketed cylinders and higher with superheated than with saturated steam. In large slow-speed pumping engines with jacketed cylinders the diagram factor is sometimes as high as 1.

218. Interchange of Heat Between Expanding Steam and Cylinder.—Let 1 2, Fig. 346, be a portion of the expansion curve of an indicator diagram. With sufficient data the dryness fractions x_1 and x_2 at the points 1 and 2 can be found as explained in Art. 212, p. 297.

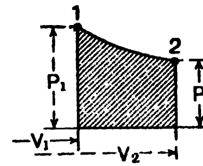


Fig. 346.

Consider 1 lb. of steam, including water, if any. If v_1 and v_2 are the specific volumes of dry saturated steam at the pressures P_1 and P_2, the volumes of the actual steam at these pressures are $V_1 = x_1v_1$ and $V_2 = x_2v_2$, the volumes of the water being neglected.

It may be assumed that the curve 1 2 is of the form $PV^n = C$. Hence $P_1V_1^n = P_2V_2^n$ and $n = \dfrac{\log P_1 - \log P_2}{\log V_2 - \log V_1}$.

If W is the work done per lb. of steam between the points 1 and 2, then $W = \dfrac{P_1V_1 - P_2V_2}{(n - 1)J}$ in heat units.

If L_1 and L_2 are the latent heats and h_1 and h_2 the sensible heats of dry saturated steam at the pressures P_1 and P_2, then the internal energies of the actual steam at the points 1 and 2 are—

$$h_1 + x_1\left(L_1 - \frac{P_1v_1}{J}\right) = E_1 \quad \text{and} \quad h_2 + x_2\left(L_2 - \frac{P_2v_2}{J}\right) = E_2$$

If Q is the heat given by the steam to the cylinder between the points 1 and 2, then $Q = E_1 - E_2 - W$.

If $E_1 - E_2 - W$ is negative, then Q is negative, and the steam gains heat from the cylinder.

On the $T\phi$ diagram the expansion line is vertical when $Q = 0$, it slopes downwards to the left when Q is positive and downwards to the right when Q is negative.

Exercises XV

1. An indicator spring is found to require an axial force of 30 lb. to shorten it 0·1 inch. If this spring is used in an indicator having a piston half of a square inch area, and a pencil mechanism which magnifies the motion of the piston six times, what would be the scale of this spring in lb. per sq. inch per inch of pencil travel?

2. A vertical two-crank compound engine has one H.P. cylinder 21 inches in diameter and one L.P. cylinder 36 inches in diameter. The stroke of both pistons is 3 feet and both piston rods are 4·5 inches in diameter and there are no tail rods. In a trial of this engine the mean effective pressures were 41·5 and 12·14 lb. per sq. inch in the H.P. and L.P. cylinders respectively. The engine speed was 100·6 revolutions per minute. From these data compute the I.H.P. of the separate cylinders.

3. An indicator diagram from a horizontal steam engine is given in Fig. 347 together with certain particulars. Determine: (*a*) The mean effective pressure;

(*b*) The indicated horse-power; (*c*) The indicated steam consumption per I.H.P. per hour. Neglect the piston rod and take the weights of 1 cubic foot of steam at the absolute pressures of 50 and 20 lb. per square inch (the pressures at A and B) as 0·1175 and 0·0498 lb. respectively. Note that the pressure scale given on the diagram is *gauge* pressure in lb. per sq. inch.

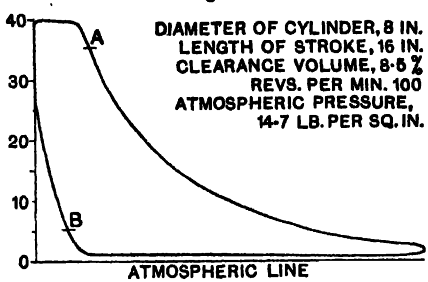


Fig. 347.

4. In a two-cylinder compound engine the diameters of the cylinders are $25\frac{1}{32}$ and $52\frac{1}{32}$ inches, and the stroke of the pistons is 5 feet. The H.P. piston rod and tail rod have a diameter of 6 inches, and the L.P. piston rod and tail rod have a diameter of 7 inches. On trial the M.E.P. in the H.P. cylinder was 53·77 lb. per sq. inch, and the M.E.P. in the L.P. cylinder was 11·69 lb. per sq. inch. Compute the M.E.P. for the whole engine referred to the L.P. cylinder.

5. A triple expansion engine has a L.P. cylinder 57 inches in diameter and a piston stroke of 39 inches. The diameter of the L.P. piston rod is 5 inches and there is no tail rod. On trial the engine developed 645 I.H.P. at 61 revs. per min. Calculate the M.E.P. referred to the L.P. cylinder.

6. The following particulars relate to a compound engine: Diameter of H.P cylinder, 28 inches. Diameter of L.P. cylinder, 57 inches. Diameter of H.P. piston rod, 6·5 inches. Diameter of L.P. piston rod, 6·25 inches. Stroke of pistons, 4 feet. There are no tail rods. Mean clearance volume 6·6 per cent. of the mean stroke volume in the case of the H P. cylinder and 6·4 per cent. in the case of the L.P. cylinder. Calculate the mean stroke volume for each cylinder, also the mean clearance volumes.

7. During a 7 hours trial of the engine of the preceding exercise the weight of steam entering the H.P. cylinder per hour was 16,162 lb. and the drainage from the intermediate receiver per hour was 297 lb. The engine speed was 80·5 revs. per min. Calculate the cylinder feed per stroke for both cylinders.

8. The co-ordinates of points on the mean diagrams for the trial of the compound engine of exercises 6 and 7 are given in the following table. The points are taken in order round the diagram in the clockwise direction. N is the number of the point, p is the absolute pressure in lb. per sq. inch, and d' is the distance of the point in inches from the left hand end of the diagram.

	H. P Diagram.		L. P. Diagram.			H. P. Diagram.		L P. Diagram.	
N	p	d'	p	d'	N	p	d'	p	d'
1	178	0·00	31·7	0·00	9	50	2·78	3·5	3·29
2	170	0·53	30	0·43	10	45	3·00	3	2·64
3	160	0·79	27	0·85	11	33	3·25	3	1·65
4	150	0·93	23	1 14	12	35	2·12	3	1·00
5	135	1·04	20	1·30	13	35	0·82	4	0·63
6	120	1·18	15	1·73	14	50	0·34	10	0 17
7	90	1·56	10	2·67	15	90	0·13	15	0·06
8	70	1·96	7	3·02	16	160	0·00	23	0·00

The extreme lengths of the diagrams are: H.P., 3·25 inches; L.P., 3·29 inches. From the above table draw the diagrams to the same pressure scale (say 1 inch to 80 lb. per sq. inch) and the same volume scale (say 1 inch to 12 cubic feet) as in Fig. 341, p. 299, and add the saturation curves.

9. Redraw the diagrams of exercise 8 eliminating the cushion steam as explained in Art. 214, p. 298.

10. Draw the temperature-entropy diagrams corresponding to the diagrams of exercise 8.

11. The cylinder diameters of a triple expansion engine are 16, 22, and 41 inches respectively, and the stroke is 27 inches. The following data are taken from the mean indicator cards:—

Cylinder	H.P.	I.P.	L.P.
Area of diagram in sq. inches	3·85	4·3	4·62
Length of diagram in inches	4·45	4·4	4·45
Scale of indicator spring (lb. per inch)	80	32	12

The admission pressure at the high-pressure cylinder is 180 lb. per sq. inch abs., and the exhaust pressure from the low-pressure cylinder is 4 lb. per sq. inch abs. The cut-off in the high-pressure cylinder takes place at 70 per cent. of the stroke. Calculate the mean effective pressure referred to the low-pressure cylinder, and the diagram factor, assuming the expansion to be hyperbolic. [U.L.]

12. The expansion curve of an indicator diagram is found to have the equation $pv^{1.1}$ = constant; determine how much heat has been taken in from the cylinder walls per lb. of steam, as the steam expands from the pressure 100 lb. per sq. inch abs. to the pressure 20 lb. per sq. inch abs.; the steam is known to be dry at the higher pressure. [C.M.S.T.]

13. At two points A and B on the expansion curve of an indicator diagram from the L.P. cylinder of a compound engine the pressures are 16 and 10 lb. per sq. inch abs., respectively, and the dryness fractions were found to be 0·795 and 0·815 respectively. Find the amount of heat taken in by the steam from the cylinder, per lb. of steam, between the points A and B.

CHAPTER XVI

CRANK EFFORT DIAGRAMS AND FLY-WHEELS

219. Crank Effort.—Referring to Fig. 348, if P is the effort on the crosshead, Q, the thrust or pull on the connecting rod, is equal to $P/\cos\phi$. At the crank pin the force Q produces a thrust or pull on the crank and a force T tangential to the path of the crank pin equal to $Q\sin(\theta+\phi)$.

Hence, $$T = \frac{P\sin(\theta+\phi)}{\cos\phi} = P(\sin\theta + \cos\theta\tan\phi).$$

If n is the ratio of the length of the connecting rod to the radius of the crank, then $\tan\phi = \dfrac{\sin\theta}{\sqrt{n^2-\sin^2\theta}}$, and therefore

$$T = P\left\{\sin\theta + \frac{\sin\theta\cos\theta}{\sqrt{n^2-\sin^2\theta}}\right\} = P\left\{\sin\theta + \frac{\sin 2\theta}{2\sqrt{n^2-\sin^2\theta}}\right\}.$$

T is called the *crank pin effort*. The moment of T about C, namely, Tr, where r is the radius of the crank, is called the *crank effort*, but as r is constant, it follows that the crank effort is proportional to T. If Cb be made equal to P, and bd be drawn parallel to AB to meet Cd at d, where Cd is perpendicular to the line of stroke, then

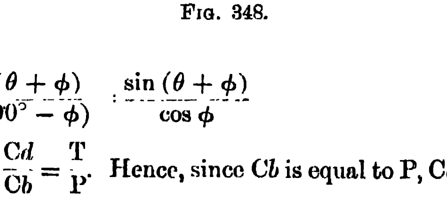


FIG. 348.

$$\frac{Cd}{Cb} = \frac{\sin(\theta+\phi)}{\sin(90^\circ-\phi)} = \frac{\sin(\theta+\phi)}{\cos\phi}$$

but $\dfrac{T}{P} = \dfrac{\sin(\theta+\phi)}{\cos\phi}$, therefore $\dfrac{Cd}{Cb} = \dfrac{T}{P}$. Hence, since Cb is equal to P, Cd must be equal to T. The construction for determining the crank effort from the piston or crosshead effort is therefore extremely simple.

220. Crank Effort Diagrams.—The construction of diagrams which shall show the crank effort for any position of the crank will be readily understood by reference to Fig. 349. In the polar curves of crank effort, the effort found by the construction or by the formula of the preceding Article is marked off, either on the crank from the centre of the crank shaft, or on the crank produced from the path of the crank pin. The

most useful crank effort diagram is the "rectangular diagram," in which the base is a straight line representing the circumference of the circle described by the crank pin, and the ordinates, perpendicular to that base, represent the crank effort.

If the base of the rectangular diagram of crank effort be made equal to the circumference of the circle described by the crank pin, then, friction being neglected, the area of the crank effort diagram for one revolution will be equal to the sum of the areas of the piston effort diagrams, but practically all that is to be learned from the rectangular crank effort diagram can be learned from it whatever be the length taken for the base.

The principal use of the rectangular crank effort diagram is to show the fluctuation of energy, which is discussed in the next Article, and for this the length of the base is immaterial.

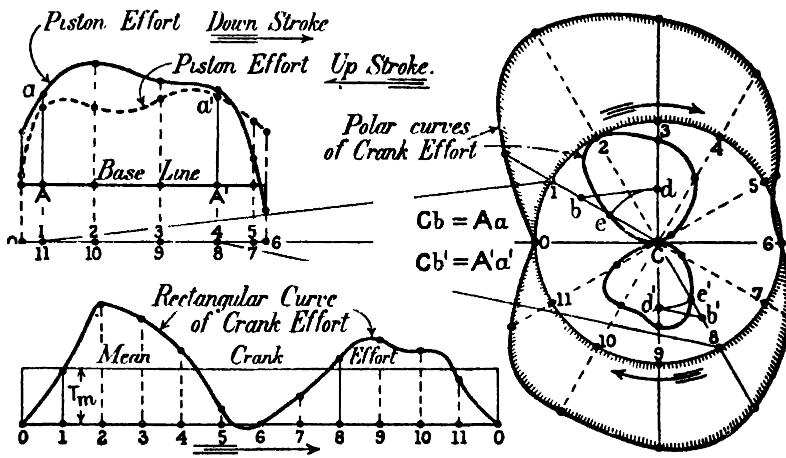


FIG. 349.

The maximum crank effort can evidently be found from either the polar or rectangular curves. The maximum crank effort is also the maximum torque on the crank shaft, and this is of great importance in designing the shaft.

If T_m is the mean effort on the crank pin and P_m is the mean effort on the piston, during one revolution, then, since the work done at the crank pin is equal to the work done on the piston in the same time, friction being neglected,

$$2\pi r T_m = 2P_m \times 2r, \text{ or } T_m = \frac{2P_m}{\pi}$$

When there are two or more cranks on a shaft, the total turning effort on the shaft at any instant is the sum of the turning efforts on the separate cranks at that instant, and the total effort may be considered as acting on any one of the cranks. Hence a diagram of total turning effort may be constructed by adding to the ordinates of the effort diagram for one crank the corresponding ordinates of the effort

diagrams for the other cranks, corresponding ordinates being those which show the efforts on the separate cranks at the same instant.

Fig. 350 shows the relative positions of three cranks on the same shaft, and Fig. 351 shows how the rectangular crank effort diagrams for these three cranks may be combined to give a total turning effort diagram. It will be observed in Fig. 351, that in order to bring the corresponding ordinates together the effort diagrams for cranks No. 2 and No. 3 have been moved forward distances corresponding to the respective angles which these cranks would have to move through to overtake No 1 crank. It is obvious that the crank effort diagrams for the separate cranks must be to the same effort scale before they can be combined into one effort diagram in the manner shown in Fig. 351. The mean total effort is of course equal to the sum of the mean efforts for the separate cranks.

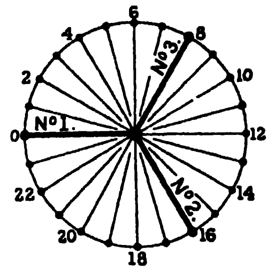


FIG. 350.

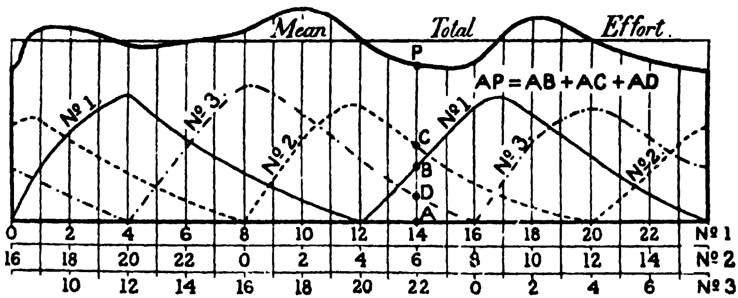


FIG. 351.

221. Fluctuation of Energy.—When the direct-acting engine mechanism is used to transmit the work done on a piston to a shaft, the turning effort on the shaft is very variable when only one crank is used, and when two or more cranks, inclined to one another, and connected to different pistons are used, the turning effort on the shaft, although much more nearly uniform, is still variable. This want of uniformity in the turning effort on the crank shaft is a characteristic of all heat engines having reciprocating pistons, and the result of this is that, except in the very improbable case in which the moment of the resistance to the turning of the shaft varies so that at every instant it is equal to the turning moment, the supply of energy to the shaft over certain intervals must be greater, while over other intervals the supply must be less than that required by the resistance.

In most cases in practice the resistance to the rotation of the crank shaft of an engine may be considered to be uniform during a complete period or cycle, and the resistance reduced to the crank pin may therefore be considered as equal to the mean effort on the crank pin during a period or cycle.

Fig. 352 shows a rectangular diagram of crank effort on a base OX, representing the path of the crank pin, and the ordinates of the line LMN represent the resistance reduced to the crank pin. In the upper part of Fig. 352, LMN is a straight line parallel to OX, while in the lower part LMN is a curved line. In each case the work done by the effort is represented by the area between the effort curve and the base, and the work done on the resistance is represented by the area between the resistance line and the base. It will be noticed that the points A, B, C, D, E, and F are the points where the effort is equal to the resistance.

Let K denote the kinetic energy in the moving parts when the crank pin is at A, then while the crank pin moves from A to B the work done by the effort is greater than that required by the resistance by the amount represented by the area a_1, and therefore the kinetic energy in the moving parts when the crank pin reaches B is $K + a_1$. Again, while the crank pin moves from B to C the work done by the

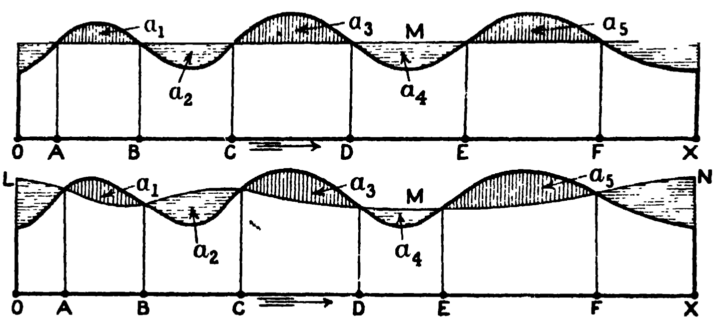


FIG. 352.

effort is less than that required by the resistance by the amount represented by the area a_2, and therefore the kinetic energy in the moving parts when the crank pin reaches C is $K + a_1 - a_2$. Similarly, the values of the kinetic energy in the moving parts when the crank pin reaches D, E, and F are, $K + a_1 - a_2 + a_3$, $K + a_1 - a_2 + a_3 - a_4$, and $K + a_1 - a_2 + a_3 - a_4 + a_5$ respectively. Between O and X the velocity of the crank pin will be a maximum at that point where the kinetic energy of the moving parts is greatest, and the velocity will be a minimum at that point where the kinetic energy is least.

The difference between the kinetic energy of the moving parts at the points of maximum and minimum speed is called the *fluctuation of energy*.

The ratio which the fluctuation of energy bears to the work done per cycle is called the *coefficient of fluctuation of energy*. In an ordinary steam engine the cycle takes place in one revolution, while in an internal combustion engine working on the Otto cycle, the cycle covers two revolutions of the crank shaft.

Referring to Fig. 352, suppose that OX represents the distance travelled by the crank pin during one cycle, and suppose that F is the point of maximum speed, and C the point of minimum speed. Let

the area between the effort curve and the base equal a, then the fluctuation of energy is represented by $a_3 - a_4 + a_5$, and the coefficient of fluctuation of energy is equal to $\frac{a_3 - a_4 + a_5}{a}$.

222. Fluctuation of Energy in Gas Engines.—In a single-cylinder, single-acting gas engine working on the "Otto cycle," the operations performed during a cycle are as follows:—

First Stroke.—The piston moves outwards, and draws in the charge of air and gas. This is the charging or suction stroke.

Second Stroke.—The piston moves inwards and compresses the charge. This is the compression stroke.

Third Stroke.—The compressed charge is ignited, an explosion takes place, and the piston is driven outwards by the expansive force of the products of combustion. This is the working stroke.

Fourth Stroke.—The piston moves inwards and expels the products of combustion. This is the exhaust stroke.

The indicator diagram is shown in Fig. 353, but the suction and exhaust pressures are shown exaggerated for the sake of clearness. Fig. 354 shows the diagram as continuous on a four-stroke base.

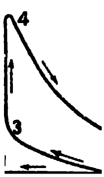


FIG. 353.

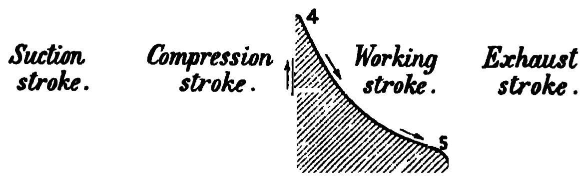


FIG. 354.

All the work delivered to the crank shaft during a cycle is delivered during the working stroke, and the work done in the cylinder during the other strokes comes from the fly-wheel.

The fluctuation of energy is obtained from the rectangular crank effort diagram as in a steam engine, but the diagram must be constructed for a complete cycle. The net work done on the useful resistance at the crank shaft and on the friction of the engine is represented by the shaded area in the working stroke in Fig. 354, minus the shaded areas in the other strokes. The maximum speed of the crank shaft is approximately at the end of the working stroke, and the minimum speed is approximately at the beginning of that stroke. Hence the fluctuation of energy is approximately equal to the work done during the working stroke, minus 1-nth of the net-work done during a cycle, where n is the number of strokes during a cycle.

If the engine is governed on the "hit or miss" principle, the governor acts by cutting off the gas, and there is no explosion and no effective work done for at least two revolutions after the completion of an effective cycle. The complete cycle then takes a number of revolutions, which is a simple multiple of two.

223. Fly-wheels.—The function of a fly-wheel is to reduce the fluctuation of speed due to the fluctuation of energy during the period

or cycle of the working of a machine. If over an interval the supply of energy to a machine is greater than the resistance requires, the moving parts increase in speed, and their kinetic energy therefore increases by an amount equal to the surplus energy; and if over another interval the supply of energy is less than the resistance requires, the moving parts decrease in speed, and their kinetic energy therefore decreases by an amount equal to the deficiency in the supply of energy. In most cases where a fly-wheel is used it is usual to neglect the kinetic energy of all the moving parts other than the fly-wheel, so that over any interval the difference between the energy supplied and the energy required is equal to the change in the kinetic energy of the fly-wheel.

If R is the radius of gyration of the fly-wheel, in feet; v the velocity, in feet per second, of a point at a distance R from the axis; ω the angular velocity in radians per second; N the speed in revolutions per minute; W the weight of the wheel in lb.; and K its kinetic energy in ft.-lb., then

$$K = \frac{Wv^2}{2g} = M_1v^2 = \frac{WR^2\omega^2}{2g} = M_2\omega^2 = \frac{W \times 4\pi^2R^2N^2}{2 \times 60^2g} = MN^2,$$

where M_1, M_2, and M are constants for a given wheel. The kinetic energy of a given wheel is therefore equal to the square of the speed, in whatever way that speed may be stated, multiplied by a constant, and for certain problems this simple rule is useful.

If I is the moment of inertia of the wheel, in lb. and foot units, then $I = WR^2$, and from this and the foregoing formulæ the following formulæ are readily deduced, namely, $K = \frac{Iv^2}{2gR^2} = \frac{I\omega^2}{2g} = \frac{2\pi^2IN^2}{60^2g}$.

It will be useful to note that for a fly-wheel rim of rectangular section the square of the radius of gyration is $\frac{r_1^2 + r_2^2}{2}$ where r_1 and r_2 (Fig. 355) are the outside and inside radii respectively. Also, the volume of the rim is $\pi(r_1^2 - r_2^2)b$, where b is the breadth of the rim.

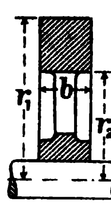


FIG. 355.

If during a period or cycle of the working of a machine the minimum and maximum speeds of the fly-wheel are N_1 and N_2 revolutions per minute respectively, then the fluctuation of energy is $\frac{2\pi^2I}{60^2g}\left(N_2^2 - N_1^2\right)$.

The difference between the maximum and minimum speeds is called the *fluctuation of speed*, and the ratio of the fluctuation of speed to the mean speed is called the *coefficient of fluctuation of speed*. If N is the mean speed in revolutions per minute, and c is the coefficient of fluctuation of speed, then $c = \frac{N_2 - N_1}{N}$. It is usual to assume that the mean speed is the arithmetical mean of the maximum and minimum speeds, so that $2N = N_2 + N_1$. Hence

$$N_2^2 - N_1^2 = (N_2 + N_1)(N_2 - N_1) = 2cN^2.$$

If U denotes the work done per period or cycle in ft.-lb., and k

denotes the coefficient of fluctuation of energy, then the fluctuation of energy is kU, and $kU = \frac{4\pi^2 WR^2 cN^2}{60^2 g} = \frac{4\pi^2 IcN^2}{60^2 g}$.

If H is the horse-power of an engine, then the work per revolution is $\frac{33000H}{N}$, where N is the speed in revolutions per minute.

The following are some values of c, the coefficient of fluctuation of speed found in practice :—

Pumps, and shearing and punching machines	0·05 to 0·03
Flour-mills	0·04 to 0·03
Looms, paper-making machines, and ordinary machine tools	0·03 to 0·025
Spinning machinery	0·02 to 0·01
Dynamos	0·007

224. Construction of Fly-wheels.—High-speed engines, that is, quick revolution engines, have fly-wheels of comparatively small diameter and are generally of the form shown in Fig. 356 or of the form shown in Fig. 357. The part D connecting the rim R and boss B or central flat L is solid and is either flat as in Fig. 356 or conical

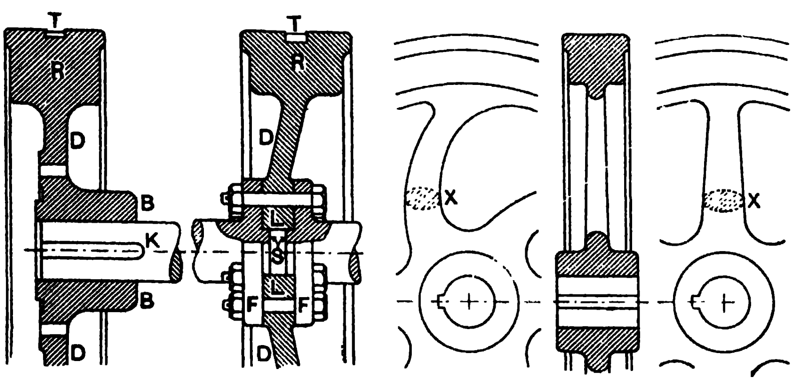


FIG. 356. FIG. 357. FIG. 358. FIG. 359. FIG. 360.

as in Fig. 357. The conical design enables the centre of gravity of the wheel to be placed nearer to the adjoining shaft bearing. When the wheel has a boss as in Fig. 356 it is bored to give a force fit on the shaft and is further secured by the key K. The wheel shown in Fig. 357 has no boss but is provided with a flat central portion L which is clamped between the flanges F of the coupling on the end of the crank shaft and is bored to fit the spigots S on the flanges, as shown, in order more readily to ensure the true running of the wheel. The recessed teeth T in the rim are for barring the engine round when necessary when not under steam.

Ordinary fly-wheels cast in one piece and having arms have generally the form shown in the section Fig. 359. The arms are either straight in side elevation as in Fig. 360 or curved as in Fig. 358. The curved arms being slightly flexible in the radial direction help to relieve the wheel of initial stresses due to unequal contraction in cooling in the

mould. The arms are generally of elliptical cross section as shown at X.

In an American design of fly-wheel, due to Mr. G. M. Hinkley, the stresses due to unequal shrinkage in cooling are to a large extent eliminated. A section of a Hinkley fly-wheel 9 feet in diameter is given in Fig. 361. The arms are more numerous than usual, twelve in this example, and they are inclined to the central plane of the wheel, passing from one side of the rim to the opposite end of the hub, alternate arms being zigzaged as shown. As first cast there is a gap

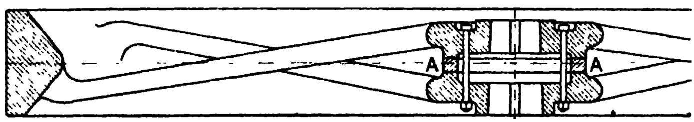


FIG. 361.—Hinkley's fly-wheel.

in the hub at AA. After the wheel has completely cooled this gap is filled by pouring in molten iron. In the original mould the gap at AA is $\frac{5}{8}$ inch wide, but the shrinkage of the rim compresses the arms and increases the gap to $1\frac{1}{2}$ inches. After filling the gap the two parts of the hub are bound together by six $\frac{5}{8}$-inch bolts and nuts, the points of the bolts being riveted over. Numerous wheels of this design have been run successfully at a rim speed of 10,000 feet per minute, which is nearly double that usually considered as a safe maximum speed for cast-iron wheels.

A common way of avoiding shrinkage stresses in wheels cast in one piece is to cast the wheel with narrow gaps A in the nave (Fig. 362). These gaps are afterwards machined and strips fitted into them and the whole nave is made sound by shrinking on steel hoops H.

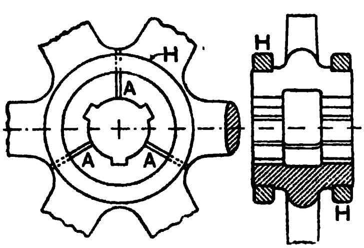


FIG. 362.

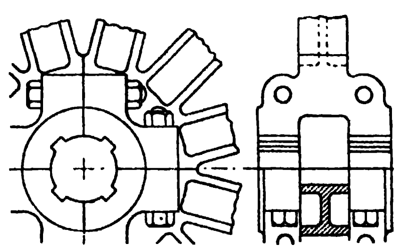


FIG. 363.

For convenience in casting and of transportation large fly-wheels, say those over 10 feet in diameter, are generally cast in two or more parts which are afterwards joined together in various ways. Usually each part consists of a portion of the rim, one or more arms and a portion of the nave. In some very large wheels the arms are cast separately.

Fig. 363 shows one form of four-part joint at the nave of a large wheel. It will be seen that eight bolts are used, four on each side. Sometimes the parts of the nave are bound together by two

steel hoops as in Fig. 362, without bolts, and sometimes both bolts and hoops are used.

Figs. 364 to 367 show various ways of uniting the rim segments together. In Fig. 364 a dowel D and cotters C are used. In Fig. 365 links L, expanded by heating, are let into recesses prepared for them, and on cooling, contract and bind the segments of the rim firmly together. A similar method is used in Fig. 366.

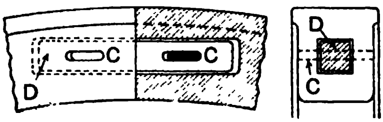


FIG. 364.

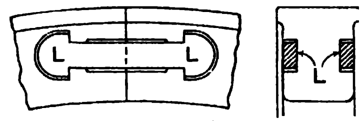


FIG. 365.

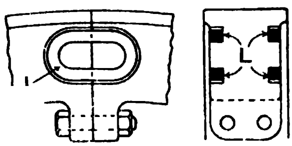


FIG. 366.

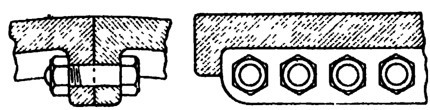


FIG. 367.

An additional fastening in the form of flanges and bolts is shown in Fig. 366. Flange and bolt joints by themselves are not recommended for deep rims such as are shown in Figs. 364, 365, and 366, but for wide rims such as are shown in Fig. 367 they are the most suitable.

The portion of the rim of a wheel between two adjacent arms is approximately in the condition of a beam fixed at the ends and loaded uniformly, the load being due to centrifugal force. If the beam is of uniform cross section the maximum bending moment occurs at the centre of such a beam and at the ends, there being no bending moment at the points which are at 21 per cent. of the span from either end. These points at which there is no bending moment are therefore the best at which to place joints such as those shown in Figs. 364 to 367. The bending moments on the rim at the arms and midway between them are of opposite sign.

Fig. 368 shows a part of a fly-wheel having bolted joints in the rim at the arms and midway between the arms, and the way in which these joints would yield is indicated. An examination of this illustration shows

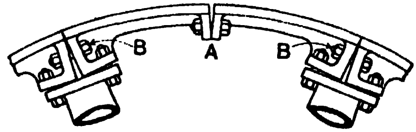


FIG. 368.

that in the joint at A between the arms the bolts should be as near as possible to the roots of the flanges. Also, it will appear that the joints at the arms are well designed to resist the bending action and that the bolts B should be as far as possible from the outside of the rim. It is not usual to have joints in the rim between the arms when there are joints at the arms.

Arms which are cast separate from the nave are frequently connected to the latter in the manner shown in Fig. 369. The inner end of each arm is turned slightly tapered and fits into a socket on the nave to which it is secured by a cotter as shown. An American

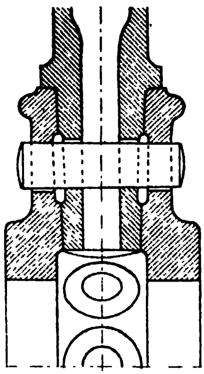


Fig. 369.

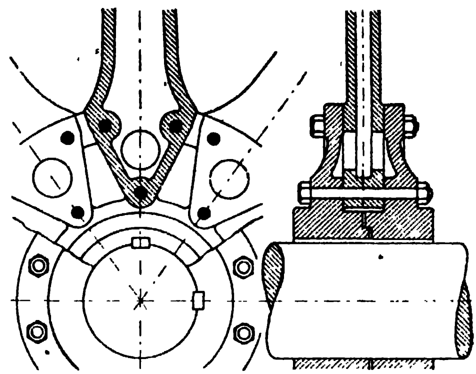


Fig. 370.

design is shown in Fig. 370. The hub is in two parts which are bolted together, clamping between them the inner ends of the arms. There are three bolts to each arm and these pass through the arm and the two parts of the hub as shown.

Exercises XVI

1. Calculate the effort on the crank pin when the crank is 45° from the inner dead centre and the effort on the piston is 5000 lb. Take the length of the connecting rod as five times the crank radius.

2. What must be the effort on the piston when the effort on the crank pin is 5000 lb. and the crank is 30° from the inner dead centre? Length of connecting rod equals five times crank radius.

3. The following particulars relate to a single-cylinder double-acting steam engine. Length of stroke, 2 feet. Length of connecting rod, 4 feet 6 inches. Clearance volume at each end of cylinder, 10 per cent. of volume swept through by piston in one stroke. Initial pressure of steam in cylinder, 80 lb. per square inch, absolute. Cut off at quarter stroke. Back pressure, 16 lb. per square inch, absolute. Assume that the steam expands after cut off according to Boyle's law.

(*a*) Construct the polar and rectangular diagrams of crank effort.

(*b*) Determine the coefficient of fluctuation of energy.

(*c*) Calculate the crank pin effort, per square inch of piston, for the point of cut off in the forward and also in the return stroke.

4. A two-cylinder engine has two cranks at right angles to one another. The particulars of the cylinders, piston effort, cranks and connecting rods are the same as for the engine of the preceding exercise. Construct the rectangular diagram of combined crank effort for this two-cylinder engine and determine the coefficient of fluctuation of energy.

5. The piston effort diagrams of a large triple-expansion marine engine are given in Fig. 371 and the relative positions of the cranks are shown in Fig. 372. The given effort diagrams show the effective effort at the crossheads after corrections for back pressure, weight and inertia of reciprocating parts, and unequal areas of pistons. There are four cylinders, one high pressure, one intermediate pressure, and two low pressure, and the diagrams and cranks for these are numbered 1, 2, 3, and 4 respectively. All the cranks have the same radius and

the length of each connecting rod is four times the crank radius. The angles between the cranks are multiples of 1–7th of 360°.

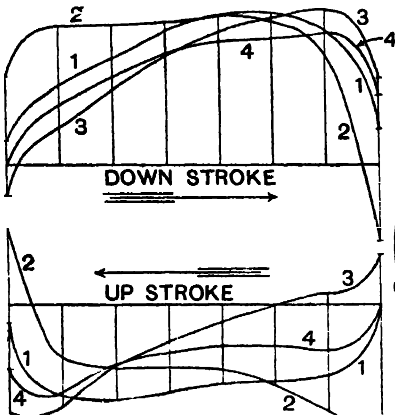


FIG. 371.

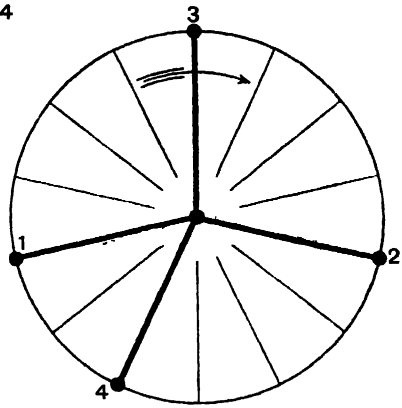


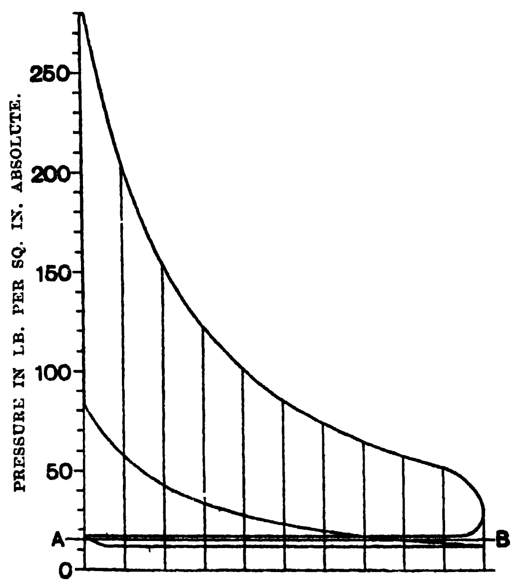
FIG. 372.

Redraw the piston effort diagrams at least twice the size of Fig. 371 and construct the crank effort diagram for each crank, then draw the combined crank effort diagram. It will be found convenient to determine the effort at crank positions which are at intervals of 1–14th of 360° as indicated in Fig. 372.

6. The indicator diagram of a horizontal gas engine is given in Fig. 373. The engine works on the Otto cycle and there is an explosion every two revolutions. There is one cylinder, 7 inches in diameter, the piston stroke is 15 inches, and the length of the connecting rod is 40 inches. The reciprocating parts weigh 64½ lb., and the mean speed of the crank shaft is 170 revolutions per minute.

Draw the rectangular crank effort diagrams and determine the coefficient of fluctuation of energy, (*a*) neglecting the inertia of the reciprocating parts, and, (*b*) taking the inertia of the reciprocating parts into account.[1]

In reproducing the indicator diagram the ordinates should be made to, say, double the scale of Fig. 373.



AB, ATMOSPHERIC LINE.

FIG. 373.

[1] See the Author's *Applied Mechanics*, Chap. XVIII.

7. The cross section of a cast iron fly-wheel is a rectangle 9 inches wide and 12 inches deep. What is the greatest outside diameter this rim can have if the safe linear velocity of the outer surface is limited to 100 feet per second and the maximum angular velocity is 250 revolutions per minute?

Taking the weight of cast iron as 450 lb. per cubic foot, calculate the kinetic energy of this rim, in ft.-lb., when it is running at its maximum speed. Also calculate the speed in revolutions per minute after the kinetic energy at the maximum speed has been reduced by 60,000 ft.-lb.

8. A fly-wheel weighing 10,000 lb. has a radius of gyration of 3·5 feet and is running at 200 revolutions per minute. What constant tangential force acting at the circumference of the wheel, which has a radius of 4·5 feet, will be required to bring it to rest in 5 revolutions?

9. The fly-wheel of a rolling mill engine is observed to change its speed from 100 revolutions per minute to 70 revolutions per minute in 5 seconds when a billet is passed into the rolls. The moment of inertia of the fly-wheel is equivalent to 30 tons at a radius of 10 feet. Find the average torque exerted on the crank shaft due to the energy drawn from the fly-wheel. [B.E.]

10. An engine of 300 horse-power has a mean speed of 160 revolutions per minute. The fluctuation of energy is 1–10th of the work done by the engine in one revolution. Taking the radius of gyration of the fly-wheel to be 3 feet, find its weight in order that the fluctuation of speed may not exceed 2 per cent. of the mean speed.

11. A gas engine working on the Otto cycle is governed by cutting out a charge from time to time. It is provided with two fly-wheels, the radius of gyration of each being $\sqrt{3 \cdot 5}$ feet. In designing the engine the following, among other, particulars were used:—Diameter of cylinder, 8 inches; stroke, 10 inches; revolutions per minute, 250; estimated mean pressures above the atmosphere—during working stroke 90, during compression stroke 20 pounds per square inch; maximum cyclical variation of speed at full load 2 per cent. Calculate the approximate weight of each wheel. Estimate also the cyclical variation of speed when there is an explosion once every eight strokes. [U.L.]

12. Determine the approximate moments of inertia and the weights of the rims for the fly-wheels of two single acting Otto cycle gas engines, one governing by throttling and the other by cutting out the gas supply.

Each engine has to develop 50 indicated horse-power at normal load, and the cyclic irregularity is not to exceed 1/50. Mean diameter of fly-wheel rim 6 feet, and revolutions per minute 180. In the throttle governing engine the explosions are 90, and in the cut-out governing engine 85 per minute.

Assume in each case that the work done on the gases by the piston during compression is 0·35 of the net work done per explosion. [U.L.]

13. In the turning moment diagram for one revolution of a steam engine the areas above and below the curve of resistance, taken in order, are + 0·53, − 0·33, + 0·38, − 0·47, + 0·18, − 0·36, + 0·35, and − 0·28 square inch.

The scales of the diagram are:—

Turning moment, 1 inch = 8000 lb.-ft.
Crank angle, 1 inch = 60°.

The mean revolutions per minute are 150, and the total fluctuation of speed must not exceed 3 per cent. of the mean. Determine a suitable cross-sectional area of the rim of the fly-wheel, assuming the total energy of the fly-wheel to be 16/15 that of the rim. The peripheral velocity of the fly-wheel is to be 50 feet per second. Take the weight of the material as 0·25 pound per cubic inch. [U.L.]

CHAPTER XVII

GOVERNORS

225. Function of a Governor.—The function of a governor is to regulate the mean speed of a machine or prime mover, or to keep the mean speed within certain limits, the limits of variation depending on the nature of the work which the machine or prime mover has to do. The limits of variation of mean speed will also depend on the sensitiveness of the governor used.

The function of the governor differs from that of the fly-wheel. The fly-wheel limits the variation of speed, during a cycle, which may be performed during a fraction of a revolution or during several revolutions, but the function of the fly-wheel is not to regulate the speed when a permanent change takes place in the load, or when the change in the load lasts for more than a cycle of operations of the machine or prime mover; this is the function of the governor, which should regulate the supply of power to the demand. For example, in a steam-engine the fly-wheel controls the variation of speed due to the difference between the effort on the crank pin and the resistance at the crank pin due to the load when the work done by the effort, during a cycle, is equal to the work done on the resistance.

A change in the average resistance should be accompanied or followed as soon as possible by a corresponding change in the average effort which is effected by the governor altering the point of cut off, or altering the initial pressure by operating a throttle valve. The governor of a reciprocating steam engine can only act during the period of admission of steam to the cylinder, and if a permanent change in the load occurs between the periods of admission, the fly-wheel exerts a controlling influence on the speed until the governor can act.

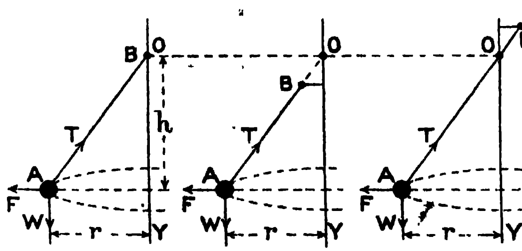


FIG. 374. FIG. 375. FIG. 376.

226. Revolving Pendulum.—In its simplest form the revolving pendulum consists of a small body A revolving about a vertical axis OY, and suspended from a point B by a thread or slender rod. In Fig. 374 the point B is on the axis OY, while in Figs. 375 and 376, B is at some fixed distance

from OY. When B is outside OY it rotates about OY with the same angular velocity as A by being on an arm fixed to a rotating spindle, of which OY is the axis. If AB is a rod, there is a joint at B which permits of the free angular movement of AB about B in the plane AOY.

As A revolves at a steady speed, AB describes the surface of a cone whose vertex is at O, where AB intersects OY, whose height is h, and whose base has a radius r. The forces acting on A in the plane AOY are, its weight W, the centrifugal force F, and the tension T in AB, and for steady motion these must balance one another. Hence, taking moments about O, $Fh = Wr$. But $F = \frac{W\omega^2 r}{g}$, where ω is the angular velocity of A about OY, therefore $\frac{W\omega^2 rh}{g} = Wr$, and $h = \frac{g}{\omega^2}$.

If g is in feet per second per second, and ω is in radians per second, then h is in feet. If A makes n revolutions per second, or N revolutions per minute, then $h = \frac{g}{4\pi^2 n^2} = \frac{60^2 g}{4\pi^2 N^2}$.

227. The Simple Conical Pendulum Governor.—One of the earliest forms of the simple conical pendulum governor, as applied to a steam

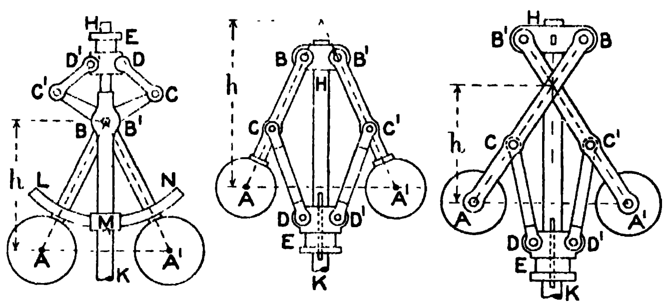


FIG. 377. FIG. 378. FIG. 379.

engine, is shown in Fig. 377. ABC and A'B'C' are the arms, jointed together and to the vertical spindle HK at BB'. Links CD and C'D' connect the arms to the sleeve E, which, while it rotates with the spindle HK, can slide up or down on it when the balls A and A' fall or rise with changes of speed. The sleeve E has a groove turned on it to receive the forked end of a lever, through which, and through other levers and links if necessary, the sliding motion of the sleeve is transmitted and converted into the motion of the throttle valve. The vertical spindle HK is driven by the engine which the governor has to control. To reduce the strain on the joint at BB' caused by the inertia of the balls when the angular velocity of the spindle changes, the arms AB and A'B' work in slots in the curved arms ML and MN, which are fixed to the spindle at M.

A later and more common form of the simple governor is that shown in Fig. 378, and to this the description just given will apply, except that the arms ML and MN are dispensed with, and the sleeve E is

driven by a key on the spindle HK, which, however, does not interfere with the vertical sliding of the sleeve on the spindle.

A modification of the design shown in Fig. 378, which makes the governor more sensitive, is that in which the axes of the joints at B and B′ are made to coincide and intersect the axis of the vertical spindle, as in Fig. 377. A still more sensitive form is that shown in Fig. 379, which is known as a *crossed arm governor*. The three designs shown in Figs. 377, 378, and 379 correspond to the three forms of the simple conical pendulum shown in Figs. 374, 375, and 376, p. 318.

Neglecting friction and the effects of the mass of the arms and sleeve, the formulæ connecting the speed with the height h for the governors described in this Article are the same as for the simple conical pendulum, namely,

$$h = \frac{g}{\omega^2} = \frac{g}{4\pi^2 n^2} = \frac{60^2 g}{4\pi^2 N^2}$$

The following results are useful in connection with calculations on governors:—

$$g = 32{\cdot}2. \qquad \sqrt{g} = 5{\cdot}6745. \qquad \frac{g}{4\pi^2} = 0{\cdot}8156.$$

$$\sqrt{\frac{g}{4\pi^2}} = 0{\cdot}9031. \qquad \frac{60^2 g}{4\pi^2} = 2936{\cdot}3. \qquad \sqrt{\frac{60^2 g}{4\pi^2}} = 54{\cdot}187.$$

228. Loaded Governors.—The simple governor is improved, particularly as regards its power of overcoming frictional resistances, by adding a central weight, which increases the downward pull on the revolving balls without increasing their centrifugal force. Fig. 380 shows a simple form of loaded governor. The central weight or load W is in the form of a disc with a central boss, which corresponds to the sleeve E in the illustrations of the preceding Article. The masses at the lower ends of the revolving arms, or pendulum weights, are in this case in the form of rollers, upon which the disc part of the central load rests, there being slots in the disc through which the revolving arms pass, as shown.

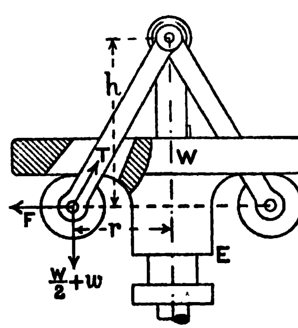


FIG. 380.

Let W equal the total weight of the central load, and w the weight of each of the pendulum weights. The centrifugal force F of each pendulum weight is equal to $\frac{w\omega^2 r}{g}$, and the downward pull on each of these weights is $\frac{W}{2} + w$, hence, taking moments about the point of suspension of the arms,

$$\left(\frac{W}{2} + w\right)r = Fh = \frac{w\omega^2 rh}{g}, \text{ and therefore, } h = \left(\frac{W + 2w}{2w}\right)\frac{g}{\omega^2}$$

Comparing this with the corresponding result for the simple governor, it is seen that for the same speed the height of this loaded governor is greater than that of the simple governor in the ratio of $W + 2w : 2w$.

More frequently the central load is suspended from the pendulum weights by links, as in Fig. 381, which shows the *Porter governor*, so called from the name of its inventor.

To determine the relation between the height and speed in the Porter governor, consider the diagram Fig. 381. Let W equal the total weight of the central load, and w the weight of each revolving ball. The central load will cause a tension in each suspension link equal to $\frac{W}{2\cos\theta}$. This tension may be resolved at the centre of each ball into a vertical component $\frac{W}{2}$, and a horizontal component Q equal to $\frac{W}{2}\tan\theta$. Taking moments about B, the point of suspension of the pendulum arms,

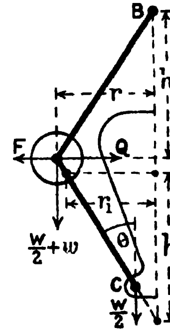


FIG. 381

$$\left(\frac{W}{2} + w\right)r + \frac{W}{2}h\tan\theta = Fh = \frac{w\omega^2 rh}{g}$$

Let $\tan\theta = \frac{r_1}{h} = \frac{qr}{h}$, then $\left(\frac{W}{2} + w\right)r + \frac{Whqr}{2h} = \frac{w\omega^2 rh}{g}$

and therefore $$h = \frac{\frac{W}{2}(1 + q) + w}{w}\frac{g}{\omega^2}$$

If $r_1 = r$, then $q = 1$,

and $$h = \frac{W + w}{w} \cdot \frac{g}{\omega^2}$$

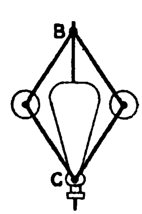


FIG. 382.

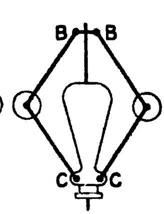


FIG. 383.

When the pendulum arms and the suspension links are of equal length, and the axes of the joints at B and C either intersect the main axis (Fig. 382) or are at equal distances from that axis (Fig. 383), then q is equal to 1. In other cases, the value of q is best found by measuring r and r_1 on a diagram to scale. It should be noted that when q is not equal to 1, its value alters as the height h changes.

229. Effect of Friction on Governors.—The frictional resistances of the various joints of the governor itself, and of the gear which the governor has to operate, may be reduced to a single force R acting on the sleeve in a direction opposite to that of its motion. When the sleeve is rising, and the speed of the governor therefore increasing, R will act downwards, and in a loaded governor this will be equivalent to altering the central load from W to W + R. Again, when the sleeve is descending R will act upwards, and this will be equivalent

to altering the central load from W to W − R. Hence for a loaded governor of the type shown in Fig. 380, $h = \frac{W \pm R + 2w}{2w} \cdot \frac{g}{\omega^2}$, the plus (+) sign being used for increasing speed, and the minus (−) sign for decreasing speed.

For the Porter governor, $h = \frac{\frac{1}{2}(W \pm R)(1 + q) + w}{w} \cdot \frac{g}{\omega^2}$ and when $q = 1$, $h = \frac{W \pm R + w}{w} \cdot \frac{g}{\omega^2}$.

The formulæ just given for the Porter governor will also apply to the simple governor when the upper joints of the suspension links are at the centres of the pendulum weights, but W will then be the weight of the sleeve. If the suspension links are jointed to the pendulum arms, as shown in Fig. 384, then $W \pm R$ must be changed to $(W \pm R)\frac{a}{l}$. The reason for the foregoing statement will be obvious from the following considerations. Draw AC parallel to A′C′. Let T′ be the tension in the suspension link when it is at A′C′, and let T be the tension in that link when it is transferred to AC. Then since the moment of T′ about B has to balance the moments of F and w about B, also since the moment of T about B has to balance the moments of F and w about B, it follows that T′a must be equal to Tl, or $T = \frac{aT'}{l}$, hence $W \pm R$ at C′ must become $(W \pm R)\frac{a}{l}$ at C.

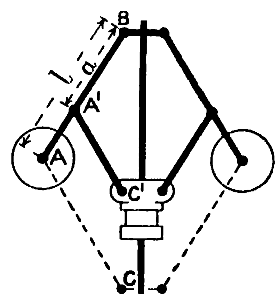


Fig. 384.

If the speed of a governor and the lift of the sleeve, or the lift of the pendulum weights, be plotted, (1) neglecting friction, and (2) taking the friction into account, instructive curves, such as are shown in Fig. 385, are obtained. HK is the lift of the sleeve. When the sleeve is at Y the speed, say in revolutions per minute, is YL when friction is neglected, YL_1 when friction is considered and the sleeve is descending, and YL_2 when friction is considered and the sleeve is ascending. Preferably the speeds are measured from a vertical axis some distance to the left of HK in order that a larger scale may be used for the speeds, and so cause the points L_1, L, and L_2 to be further apart. The abscissæ and ordinates of the curve ALB represent the speed and lift respectively when friction is neglected. The abscissæ and ordinates of the curve $A_1L_1B_1$ represent the speed and lift respectively when friction is considered and the sleeve is descending. Lastly, the abscissæ and ordinates of the curve $A_2L_2B_2$

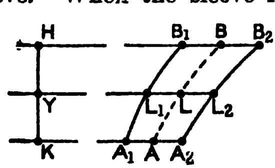


Fig. 385.

represent the speed and lift respectively when friction is considered and the sleeve is ascending.

230. Graphics of Forces Acting on a Governor.—The relations between the various forces acting on a governor are readily determined graphically and the graphical solution of many governor problems is to be preferred to the analytical.

As an example of the graphical method take the case of the Porter governor. The outline of one half of such a governor is shown in Fig. 386. The forces acting at the centre of the revolving ball are its weight w, the centrifugal force F, and the tensions in the upper and lower links. Neglecting friction, the vertical component of the tension in the lower link is equal to half the central load W, hence, if the right-angled triangle ABC (Fig. 387) be drawn, AB being vertical and made equal to $\frac{1}{2}$W to a selected scale, and BC parallel to the lower link, then BC will represent the tension in the lower link. The polygon of forces CDEB for the forces acting on the ball may now be drawn, BE being vertical and equal to w, CD parallel to the upper link and ED

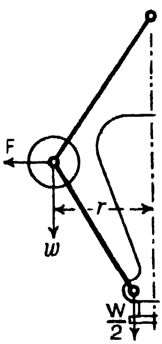


FIG. 386.

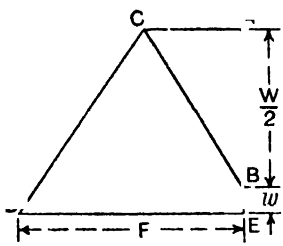


FIG. 387.

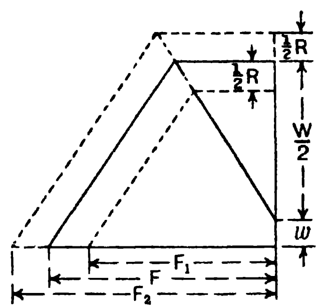


FIG. 388.

horizontal. If W and w are known, then, by the above construction, the centrifugal force F, for any given configuration of the governor, may be found and from the force F the speed may be determined since $F = \frac{w\omega^2 r}{g} = \frac{wr}{g}\left(\frac{\pi N}{30}\right)^2$.

Hence $\omega = \sqrt{\frac{Fg}{wr}}$ and $N = \frac{30}{\pi}\sqrt{\frac{Fg}{wr}}$, r of course being measured from the drawing giving the configuration of the governor.

The effect of friction, represented for the whole governor by a force R at the sleeve, is allowed for as shown in Fig. 388. F_1 is the centrifugal force when the sleeve is descending and F_2 is the centrifugal force when the sleeve is ascending at the time when the configuration of the governor is as shown in Fig. 386.

As another example, a governor in which the links carrying the sleeve are jointed to the ball arms at intermediate points, as in Fig 389, may be taken. First neglecting friction, BC (Fig. 390) the tension in each suspension link is determined as in the previous example. Next replace the pull on the arm through the suspension

link, by a force P parallel to the link, acting at the centre of the ball and having a magnitude BC′ equal to $\frac{a}{l}$. BC. A construction for finding C′ is shown. On a straight line through B make BH $= l$ and BK $= a$. Join HC and draw KC′ parallel to HC. BEDC′ the polygon of forces acting at the centre of the ball may now be drawn and F the centrifugal force determined.

The constructions when friction is considered are clearly shown in Fig. 390.

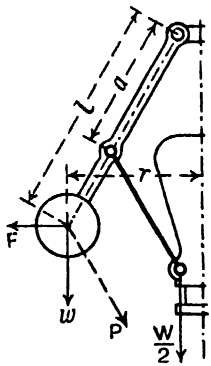


FIG. 389.

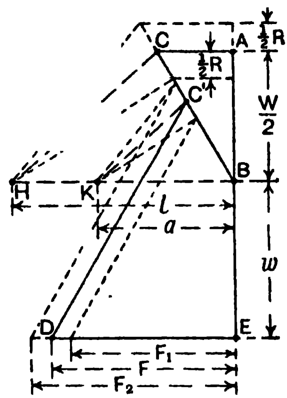


FIG. 390.

The simplicity of the graphic method is further illustrated by its application to the *Proell* governor, shown in Fig. 391, which resembles the governor of the preceding example inverted The tension in the upper link is such that its vertical component is equal to $\frac{W}{2} + w$, where as before W is the central load and w the weight of one ball. Hence, if the right-angled triangle AEC (Fig. 392) be drawn, AE being vertical and equal to $\frac{W}{2} + w$, and EC parallel to the upper link, then EC is the tension in the link. As in the preceding example the force exerted through the link on the ball arm is replaced by a force P acting at the centre of the ball and parallel to the link. The magnitude of P is equal to $EC' = EC \times \frac{a}{l}$. It will be observed that the length a varies with the configuration of the governor on account of the bent form of the ball arm. The polygon of forces now acting at the centre of the ball is completed by drawing BD horizontal and C′D parallel to the line joining the centre of the ball and the centre of the bottom joint of the ball arm.

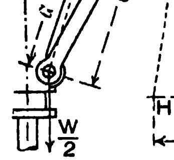


FIG. 391.

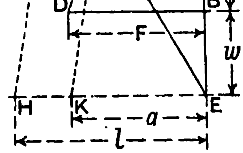


FIG. 392.

Friction is readily allowed for as in the preceding example.

231. Effect of Masses of Revolving Arms and Links.—In the preceding Articles the effect on the equilibrium of a governor of the masses of its revolving arms and links has been neglected, but in practical cases these masses must be taken into account.

A rough rule often used is to add to the weight of each ball one half the weight of any arms or links connected to it and then proceed with the ball whose weight has been so increased, as when the masses of the arms and links are neglected.

A more accurate rule is to take the effective weight of the ball as its own weight plus *one half* the weight of the arms and links connected directly to it *when considering the gravity effect of the ball*, but to take the weight of the ball as its own weight plus *one third* of the weight of the arms and links connected directly to it *when considering the centrifugal force and speed of the ball.* The arms and links are assumed to be approximately uniform in cross section.

When there is a central load, as in a Porter governor, one half of the weight of the links connected to it should be added to the central load.

It may be pointed out here that the resultant centrifugal force of an arm or link is not the centrifugal force of its mass, supposed concentrated at its centre of gravity. This is only true for a body which may be divided into a number of thin plates the centres of gravity of which all lie on a line parallel to the axis of revolution.[1]

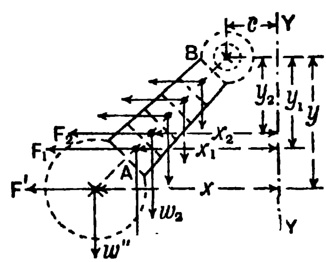


FIG. 393.

The case of an arm or link whose cross section is not uniform may be treated as follows. Divide the arm AB (Fig. 393) into a number of parts by parallel sections and compute their weights w_1, w_2, etc. Assume that the weights w_1, w_2, etc., and the centrifugal forces F_1, F_2, etc., of these parts act through the points on the axis of the arm which are midway between the parallel sections. YY is the axis about which the arm revolves and ω is the angular velocity of the arm about YY.

$$F_1 = \frac{w_1\omega^2 x_1}{g}, \quad F_2 = \frac{w_2\omega^2 x_2}{g}, \text{ and so on.}$$

Let w' be the weight which, concentrated at the centre of the ball, will have the same centrifugal effect F' as that of the arm.

Taking moments about the axis of the pin at the end B of the arm,

$$F'y = F_1 y_1 + F_2 y_2 + \text{etc.}$$

Therefore, $$\frac{w'\omega^2 x}{g}y = \frac{w_1\omega^2 x_1}{g}y_1 + \frac{w_2\omega^2 x_2}{g}y_2 + \text{etc.}$$

Hence, $$w' = \frac{w_1 x_1 y_1 + w_2 x_2 y_2 + \text{etc.}}{xy}$$

The value of w' varies with the inclination of the arm, and therefore in determining its value the arm should be placed in its mean position.

[1] See the Author's "Applied Mechanics," p. 20.

Let w'' be the weight which concentrated at the centre of the ball will have the same gravity effect as that of the arm.

Taking moments about the axis of the pin,

$$w''(x - c) = w_1(x_1 - c) + w_2(x_2 - c) + \text{etc.}$$

$$\text{Hence, } w'' = \frac{w_1(x_1 - c) + w_2(x_2 - c) + \text{etc.}}{x - c}$$

232. Controlling Force.—When a governor is running steadily in a definite position or with a definite configuration the forces acting on it are in equilibrium. Considering one revolving ball the centrifugal force on this ball tends to make it move further out while other forces which may be due to gravity or the action of springs tend to move the ball inwards. If all the forces which tend to move the ball inwards be replaced by a single equivalent force, acting on the ball in an inward radial direction, this force is called the *controlling force* on the ball.

At a steady speed the controlling force is equal in magnitude but opposite in direction to the centrifugal force. The controlling force is different for different configurations of the governor and it may be determined by the principles of statics as explained in preceding Articles, or it may be found experimentally by measuring the outward radial forces necessary at the balls to produce equilibrium for any given configuration, the controlling forces being equal and opposite to these. There are of course two extreme values of the controlling force for any given configuration of the governor, when friction is considered, one for the case where the given configuration is reached by the movement of the sleeve in one direction and the other where it is reached by the movement of the sleeve in the other direction. The actual controlling force when the governor is working may not be the same as when it is at rest because the friction of the various joints when in motion is generally less than when they are at rest.

233. Sensitiveness of Governors.—The greater the change in the level of the revolving balls of a governor for a given *percentage* or fractional change in speed the greater is its sensitiveness, and the sensitiveness may be defined as the change in level of the revolving balls, due to a change of speed of, say, 1 per cent.

Consider the case where the axis of the top joint intersects the main axis (Figs. 377, 380, 381, and 382), and let the friction be neglected. In this case the change in the level of the revolving balls is the same as the change in the height h of the governor. If n is the speed of the governor in revolutions per second when the height is h, then for the simple governor and also for the loaded governor h and n are connected by an equation of the form $h = \frac{c}{n^2}$, where c is a constant depending on the type of governor and the various weights. Also, when friction is considered, c has one value for increasing speeds, and another value for decreasing speeds. Let the speed increase from n to xn. (If the increase in speed is 1 per cent., $x = 1{\cdot}01$.) The height h will decrease to h_1

where $h_1 = \frac{c}{x^2n^2} = \frac{h}{x^2}$. Hence, $h - h_1 = \Delta h = h - \frac{h}{x^2} = h\left(\frac{x^2 - 1}{x^2}\right)$.

This shows that the sensitiveness of the governor is directly proportional to the height h, and it follows from this investigation that *the sensitiveness of the loaded governor is the same as that of the simple governor when friction is neglected.*

Many writers of note state that, friction being neglected, the loaded governor is more sensitive than the unloaded governor, and it is therefore necessary that this point should be considered more fully. Take a simple governor in which the revolving balls each have a weight w, and let this governor be converted into a loaded governor, say of the Porter type, by adding a central load of weight W, and for simplicity let the factor q (Art. 228) equal 1; then for the unloaded governor $h = \frac{g}{4\pi^2 n^2}$, and for the loaded governor $h = \frac{W + w}{w} \cdot \frac{g}{4\pi^2 n^2}$. Now if these governors are run at the *same speed*, the height of the loaded governor will be $\frac{W + w}{w}$ times the height of the unloaded governor, and under these circumstances the loaded governor would be $\frac{W + w}{w}$ times as sensitive as the unloaded governor; but what really happens in practice is that when the simple governor is replaced by a loaded governor the height h is about the same for both, and consequently the loaded governor is run about $\sqrt{\frac{W + w}{w}}$ times as fast as the unloaded governor, and the one governor is then no more sensitive than the other when friction is neglected.

Consider now the effect of friction on the sensitiveness of the governor. For the Porter governor, in its simplest form, it has been shown that $n^2 = \frac{W \pm R + w}{w} \cdot \frac{g}{4\pi^2 h}$, where R is the force required at the sleeve to overcome the friction of the governor and the gear which it has to operate. For a given value of h there are evidently two values of n, namely,

$$n_1 = \sqrt{\frac{W - R + w}{w} \cdot \frac{g}{4\pi^2 h}}, \text{ and } n_2 = \sqrt{\frac{W + R + w}{w} \cdot \frac{g}{4\pi^2 h}}.$$

Referring to the diagram Fig. 385, if n_1 is represented by the point L_1, then n_2 is represented by the point L_2. If the sleeve is at the level Y, it must have reached that level either by falling from a higher level, or by rising from a lower level. Suppose that the sleeve reached the level Y by falling from a higher level, due to a diminution in speed, then its speed must be n_1. Now suppose that the speed diminishes still further, the sleeve will again fall, but *the friction will not affect the sensitiveness of the governor;* the sensitiveness will be simply proportional to h, as has already been shown. Next, suppose that instead of the speed diminishing to less than n_1 it begins to increase after coming down to n_1, then there can be no change in the level of the sleeve until the speed has increased to n_2. If after the speed has increased to n_2 it goes on increasing, the sleeve will continue to rise, and *the sensitiveness will again be unaffected by the friction.*

If n is the mean speed $= \frac{1}{2}(n_1 + n_2)$, and represented by the point L (Fig. 385), then $\frac{n_2 - n_1}{n}$ is the coefficient of fluctuation of speed of the governor *when the direction of the motion of the sleeve is reversed*, and the smaller this coefficient is, the more sensitive is the governor.

The value of the expression $\frac{n_2 - n_1}{n}$ for a Porter governor of the simplest form is found as follows:—

$$n_2 = \sqrt{\frac{W + R + w}{w} \cdot \frac{g}{4\pi^2 h}}, \quad n_1 = \sqrt{\frac{W - R + w}{w} \cdot \frac{g}{4\pi^2 h}}, \text{ and}$$

$$n = \sqrt{\frac{W + w}{w} \cdot \frac{g}{4\pi^2 h}} \text{ (see footnote), hence}$$

$$\frac{n_2 - n_1}{n} = \frac{\sqrt{W + R + w} - \sqrt{W - R + w}}{\sqrt{W + w}}$$

$$\sqrt{1 + \frac{R}{W + w}} - \sqrt{1 - \frac{R}{W + w}}$$

If W be increased, the term $\sqrt{1 + \frac{R}{W + w}}$ decreases, and the term $\sqrt{1 - \frac{R}{W + w}}$ increases, therefore the value of $\frac{n_2 - n_1}{n}$ decreases as W increases. Hence, considering the effect of friction on a loaded governor, the sensitiveness is greater the heavier the central load, and consequently the loaded governor is more sensitive than the unloaded governor when the pendulum weights are the same in both. But the unloaded governor may be made as sensitive as the loaded governor by increasing the pendulum weights. Let w_1 = weight of each ball of a simple or unloaded governor, w = weight of each ball of a loaded governor, and W = weight of central load. Then for the unloaded governor W = 0, and

$$\frac{n_2 - n_1}{n} = \sqrt{1 + \frac{R}{w_1}} - \sqrt{1 - \frac{R}{w_1}}$$

For the loaded governor

$$\frac{n_2 - n_1}{n} = \sqrt{1 + \frac{R}{W + w}} - \sqrt{1 - \frac{R}{W + w}}$$

Hence if $\frac{n_2 - n_1}{n}$ is the same for both governors, $w_1 = W + w$.

The sensitiveness of a governor is defined by numerous writers as the ratio of the difference between the maximum and minimum speeds between which the governor works to the mean speed, and is given by

NOTE.—The mean of the rising and falling speeds for a given level of sleeve, and for a given value of R, is not quite the same as the speed for the same level when R = 0, but as R is generally small compared with W + w, the error introduced by taking n as above may be neglected.

the expression $\frac{N_1 - N_2}{N}$ where N_1 is the maximum speed, N_2 the minimum speed, and N the mean speed.

234. Stability of a Governor.—If for a definite steady speed the path of the revolving balls has a definite radius, friction being neglected, and any increase of speed causes that radius to increase, the governor is said to be *stable.* Stated in another way, suppose that while a governor is running at a steady speed the radius of the path of the balls is altered by the application of an external force, say on the sleeve, then if when that external force is removed the balls return to their former position (except in so far as they are prevented from doing so by friction) the governor is said to be stable.

235. Isochronism—Hunting.—A governor is said to be *isochronous* when for a particular speed the revolving balls will remain in any position within the range of the governor. The slightest change from the particular speed will cause such a governor to take up one of its extreme positions, the outer position for an increase of speed and the inner for a decrease. Such a governor would evidently be over sensitive and of no practical use because it would be continually oscillating between its extreme positions, alternately giving full steam and none at all. This oscillation of a governor due to over sensitiveness is known as *hunting.*

236. Parabolic and Crossed-Arm Governors.—Before discussing the *parabolic governor* it will be useful to note a few properties of the parabola. Referring to Fig. 394 the curve AQP is part of a parabola

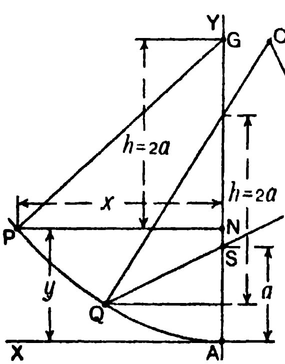


FIG. 394.

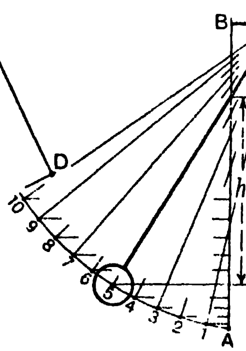


FIG. 395.

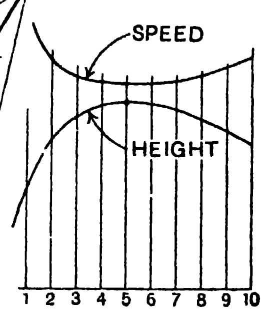


FIG. 396.

of which A is the vertex, S the focus, and AY the axis. P being any point on the curve, PN is a perpendicular from P to AY. If PN = x, and AN = y, then $x^2 = 4ay$, where a = AS. If PG the normal to the curve at P meets the axis at G then NG is called the sub-normal and this is constant and equal to $2a$.

The centre of curvature of the parabola at any point Q is found as follows. Draw the normal QC to the curve at Q. Join QS and produce it to D making SD = QS. Draw DC perpendicular to QD to meet QC at C. C is the centre of curvature required and a circle with

centre C and radius CQ will more nearly agree with the parabola at Q than any other circle which may be drawn.

If the path which the centre of a governor ball has, in the plane containing the axis of the governor and the centre of the ball, be a parabola such as AQP whose axis AY is the axis of the governor, then for any position P of the ball the virtual arm is the normal PG and the height h of the conical pendulum is constant. Hence the steady speed for equilibrium of such a governor will be the same for any position of the ball on the parabola. Such a governor would therefore be isochronous. As has already been pointed out the isochronous governor is impracticable, but by choosing a curve which differs slightly from a parabola for a given range, a workable sensitive governor may be obtained.

The simplest form of governor approximating to the parabolic governor is the *crossed-arm governor.* One arm of such a governor is shown in Fig. 395, the centre of suspension C and the revolving ball being on opposite sides of the axis AB. In designing a crossed-arm governor it is instructive to plot the height h for different positions of the arm as shown in Fig. 396. It will be seen that from the lowest position up to position 5 the height h increases and beyond this h diminishes. The corresponding speed curve is also plotted in Fig. 396.

It is obvious that the lowest practicable position for this governor is that marked 5 because in descending below this the speed for equilibrium would have to increase.

237. Effort of a Governor.—By the *effort* of a governor is meant the force which it is capable of exerting at the sleeve for a given percentage or fractional change of speed.

Consider first the effort of the Porter governor for which $\omega^2 = \frac{W + w}{w} \cdot \frac{g}{h}$. Let the speed increase from ω to $x\omega$, and let a force Q be applied at the sleeve in a downward direction, Q being just sufficient to prevent the sleeve rising. This will evidently be equivalent to increasing the central load from W to W + Q.

$$\text{Then, } x^2\omega^2 = \frac{W + Q + w}{w} \cdot \frac{g}{h}. \qquad \text{But, } \omega^2 = \frac{W + w}{w} \cdot \frac{g}{h}.$$

$$\text{Therefore, } x^2 \cdot \frac{W + w}{w} \cdot \frac{g}{h} = \frac{W + Q + w}{w} \cdot \frac{g}{h}.$$

$$\text{Hence, } Q = (W + w)(x^2 - 1).$$

If the force Q be gradually diminished to zero, the sleeve will rise until $h = \frac{W + w}{w} \cdot \frac{g}{x^2\omega^2}$, and the average value of the effort on the sleeve during the rise will be $\frac{1}{2}Q$, or $\frac{1}{2}(W + w)(x^2 - 1) = P$, and this is the resistance at the sleeve which this governor is capable of overcoming with an increase of speed from ω to $x\omega$. For a decrease in speed from ω to $x\omega$ it follows that P, now acting in the opposite direction, is equal to $\frac{1}{2}(W + w)(1 - x^2)$. For a change of speed of 1 per cent. $P = 0{\cdot}01(W + w)$.

Converting the Porter governor into an unloaded governor by removing the central load or making W = 0, it follows from the fore-

going proof that $P = \frac{1}{2}w(x^2 - 1)$, or $\frac{1}{2}w(1 - x^2) = 0{\cdot}01w$, for a change of speed of 1 per cent. If in the unloaded governor the sleeve is suspended, as in Fig. 384, then P, as just given, must be increased in the ratio of $l : a$, supposing that the suspension links and the pendulum arms are equally inclined in the main axis.

It is evident that in order that the unloaded governor may exert the same effort as the loaded governor of the Porter type, $\frac{l}{a}w_1 = W + w$, where w_1 is the weight of each ball of the unloaded governor.

For the loaded governor of the type shown in Fig. 380,

$$P = \tfrac{1}{2}(W + 2w)(x^2 - 1)$$

It will help to the understanding of what is meant by the effort of a governor if the diagram of effort on the sleeve and the velocity diagram of the governor be studied together. Two cases, which are illustrated by Figs. 397 and 398, will be considered.

In the first case (Fig 397) the effort on the sleeve is assumed to be uniform and equal to Q while the sleeve rises from the level X to the level Y. The speed curves are plotted as described in Art. 229, p. 322. Assuming that when the sleeve is at the level X the governor

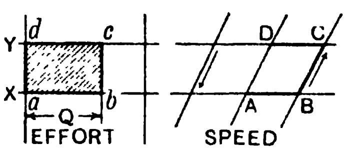


FIG. 397.

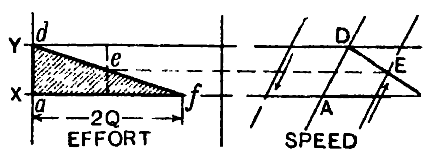


FIG. 398.

is running freely, there being no force on the sleeve due to the valve gear which is operated by the governor, then the speed of the governor will be represented by the point A. Suppose now that there is a decrease in the load on the engine which requires that the sleeve be raised to the level Y. After this change of level has been effected, and the valve gear ceases to cause any force to be exerted on the sleeve, the free steady speed of the governor will then be represented by the point D.

The changes in speed and effort during the performance of the above operation are: (1) The increase of speed represented by AB before the sleeve begins to rise, the effort meantime increasing from zero to *ab*. (2) A further increase in speed during the upward travel of the sleeve, the speed at the end of this travel being represented by the point C, and the effort remaining constant the line *bc* is vertical. (3) The decrease in speed represented by CD after the sleeve has been raised to the level Y while the effort diminishes to zero as shown by the horizontal line *cd*. The work done in raising the sleeve against the resistance offered by the valve gear is represented by the area of the rectangle *abcd*.

In the second case (Fig. 398) the effort is assumed to be equal to 2Q at the beginning of the upward travel of the sleeve and then to

diminish at a uniform rate to zero while the sleeve is rising to the level Y. The changes in the speed and effort during the operation are: (1) The increase in speed represented by AF before the sleeve begins to rise while the effort increases from zero to 2Q represented by *af*. (2) The continuous decrease in speed during the upward travel of the sleeve represented by the curve FED while the effort diminishes from *af* to zero as shown by the sloping line *fed*. The work done by the varying effort is the same as in the first case and is represented by the area of the triangle *afd*.

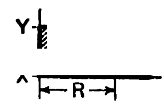


FIG. 399.

In practice the form of the effort diagram probably approximates to that shown in Fig. 399 where R is the mean effort during the operation of changing the level of the sleeve.

238. Power of a Governor.—By the *power* of a governor is meant the amount of work which it is capable of doing at the sleeve for a given percentage or fractional change of speed. The work done at the sleeve is equal to the mean effort of the governor multiplied by the distance through which the sleeve moves for the given change of speed. Thus if P = mean effort, k = lift of sleeve, and U = the power of the governor, then $U = Pk$.

For the Porter governor (Fig. 382), $P = \frac{1}{2}(W + w)(x^2 - 1)$,

$$\Delta h = \left(\frac{x^2 - 1}{x^2}\right)h, \quad k = 2\Delta h = 2\left(\frac{x^2 - 1}{x^2}\right)h,$$

$$U = Pk = (W + w)\left(\frac{x^2 - 1}{x}\right)^2 h.$$

For the direct loaded governor (Fig. 380), $P = \frac{1}{2}(W + 2w)(x^2 - 1)$,

$$\Delta h = \left(\frac{x^2 - 1}{x^2}\right)h = k, \quad U = Pk = \tfrac{1}{2}(W + 2w)\left(\frac{x^2 - 1}{x}\right)^2 h.$$

For the simple governor (Fig. 384), except that B and C′ are on the main axis,

$$P = \tfrac{1}{2}w(x^2 - 1)\frac{l}{a}, \quad \Delta h = \left(\frac{x^2 - 1}{x^2}\right)h,$$

$$k = 2\Delta h\frac{a}{l} = 2\left(\frac{x^2 - 1}{x^2}\right)\frac{ha}{l}, \quad U = Pk = w\left(\frac{x^2 - 1}{x}\right)^2 h.$$

For a change in speed of 1 per cent. $\left(\frac{x^2 - 1}{x}\right)^2 = 0{\cdot}0004.$

For a change in speed of 10 per cent $\left(\frac{x^2 - 1}{x}\right)^2 = 0{\cdot}0364.$

Most writers define the *power of a governor* as the work done by the controlling force while the balls move over their extreme radial range of travel. It should however be clearly understood that this work done by the controlling force is not the work available for overcoming the resistance due to the valve gear operated by the governor.

Further reference to the power of a governor will be found in Art. 242, p. 336.

239. Helical Springs for Governors. The springs used in spring-loaded governors are most commonly of the helical form made of round steel wire (Fig. 400) but helical springs made of square steel (Fig. 401) are also used. The springs may be in tension (Fig. 400) or in compression (Fig. 401). The *deflection* δ of the spring is directly proportional to the load P. The *stiffness* S of the spring is the load required to deflect it 1 inch, when the spring is initially unloaded, or it is the increase in the load to increase the deflection 1 inch.

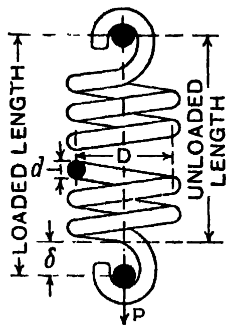


FIG. 400.

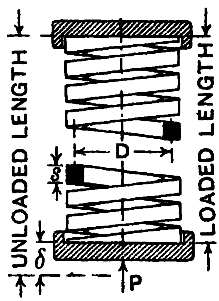


FIG. 401.

The following formulæ may be used in the preliminary design of helical springs for governors.

$$\delta = \frac{nPD^3}{1{,}500{,}000d^4}, \quad S = \frac{1{,}500{,}000d^4}{nD^3}, \text{ for round section.}$$

$$\delta = \frac{nPD^3}{2{,}150{,}000s^4}, \quad S = \frac{2{,}150{,}000s^4}{nD^3}, \text{ for square section.}$$

number of coils in spring.

$$\text{Greatest safe load } = \frac{20{,}000d^3}{D} \text{ for round section.}$$

$$\text{Greatest safe load } = \frac{27{,}000s^3}{D} \text{ for square section.}$$

Dimensions in inches and forces in pounds.

240. Spring-Loaded Governors.—A well known form of spring-loaded governor is that designed by Mr. Wilson Hartnell of Leeds. An example of this type of governor is shown in Fig. 402. Two bell-crank levers L each carry a ball B at the end of one arm and a roller R at the end of the other. Each lever is mounted on a pin I carried by a frame A which is attached to the spindle S. The centrifugal forces of the balls cause the rollers R to press against the collar C on the sleeve E. The upward pressure of the rollers on the collar of the sleeve is balanced by the downward thrust of the helical spring H, which is in compression.

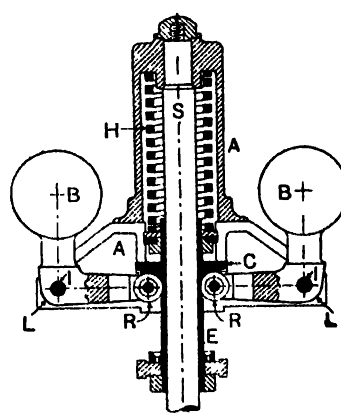
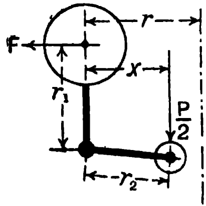


FIG. 402

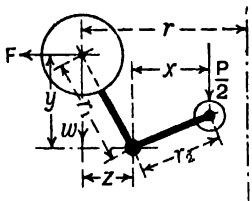
The angle of the bell-crank levers in Fig. 402 is 90° but in practice it is generally rather greater.

Taking moments about the axis of the pin upon which a bell-crank lever is mounted the equations of equilibrium are stated under Figs. 403, 404, and 405, which show three representative positions of one



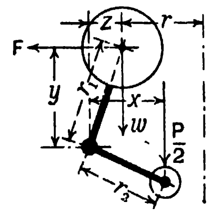
$\mathrm{F}r_1 = \frac{\mathrm{P}}{2}x.$

FIG. 403.



$\mathrm{F}y + wz = \frac{\mathrm{P}}{2}x.$

FIG. 404.



$\mathrm{F}y - wz = \frac{\mathrm{P}}{2}x.$

FIG. 405.

lever. F is the centrifugal force of one ball the weight of which is w, and P is the total thrust exerted by the spring. The gravity moment wz of a ball is small compared with the moment of the centrifugal force and if wz is neglected the equation of equilibrium reduces to $\mathrm{F}y = \frac{\mathrm{P}}{2}x$. If the bell crank angle is 90° and the gravity moment is neglected then the equation of equilibrium is $\mathrm{F}r_1 = \frac{\mathrm{P}}{2}x$ for any position.

Another example of a spring-loaded governor is shown in Fig. 406.

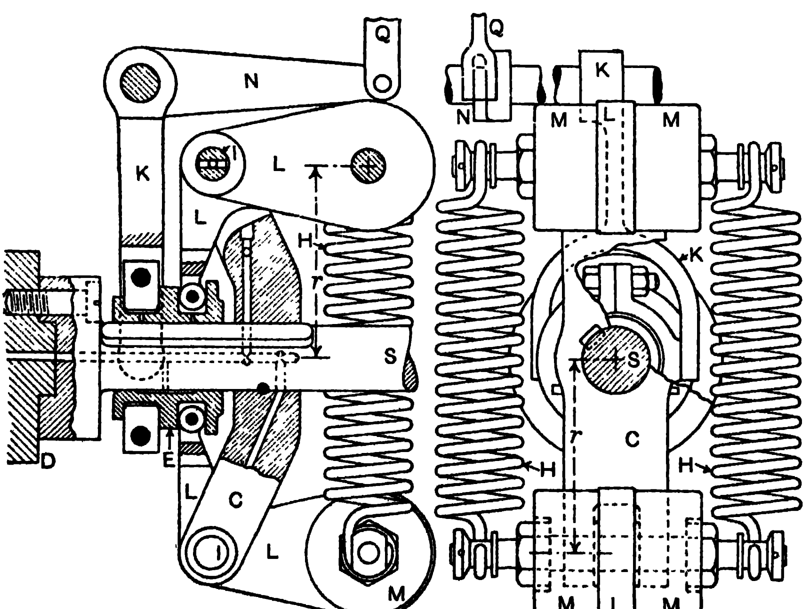


FIG. 406.

This is the governor of the Allen high-speed triple-expansion engine illustrated on p. 237. The spindle S is attached directly to one end D of the crank shaft. Two bell-crank levers L are mounted on pins I fixed in the carrier C which is secured to the spindle S. The centrifugal forces of the masses M and the crank arms carrying them are balanced by the pull exerted by the two helical springs H. The movement of the bell-crank levers is communicated to the sleeve E which in turn actuates the levers K and N to the latter of which is jointed the rod Q which is connected to the throttle valve lever. Details of the throttle valve are given on p. 269. Oil to the various joints and rubbing surfaces is supplied through the passages shown by the system of forced lubrication.

If P is the total pull of the springs when their length is $2r$ (Fig. 407), l the length of the springs when free or unloaded (Fig. 408), and S the stiffness of each spring, then the deflection δ of the springs when loaded is $2r - l$. Since the deflection of a spring is proportional to the load which it carries it follows that $P:2S::2r-l:1$. Hence, $P=2S(2r-l)=F$. Now $2S(2r-l)$ is of the form $ar - b$, where a and b are constants for a particular spring. The equation $P = ar - b = F$ is the equation to a straight line whose ordinates are the controlling force P and abscissæ the radius r.

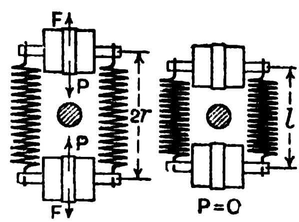


FIG. 407. FIG. 408.

241. Adjustable Supplementary Load for Governors.—Spring-loaded governors are generally provided with a supplementary spring connected to the linkwork between the sleeve and the valve in such a way as to add to the load on the sleeve due to the main spring or springs. Also, it is arranged that the length of this supplementary spring and therefore the force exerted by it may be altered by hand while the engine is running. Altering the length of the supplementary spring will of course alter the total load on the sleeve and will therefore change the speed at which the governor will be in equilibrium in any given position, an increase in the load causing an increase in the speed.

In Fig. 409, H is a supplementary spring coupled to the lever L which transmits the motion of the sleeve E upon which the main spring acts directly. The length of the spring H may be adjusted by means of the screw S and nut N.

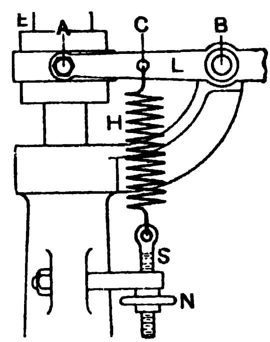


FIG. 409.

In considering the equilibrium of a governor provided with a supplementary spring the force exerted by this spring must be first reduced to the line of action of the resultant force exerted by the main spring or springs and then added to that resultant force. For example, in Fig. 409 if AB is twice BC then the force on the

sleeve due to the supplementary spring is half the force exerted by that spring at C.

242. Curves of Controlling Force.—Plotting values of the controlling force as ordinates and corresponding values of the radius of the ball path as abscissæ a curve is obtained forming the upper boundary line of a diagram which exhibits the sensibility, stability, and power of the governor. This form of governor diagram is due to Mr. Wilson Hartnell.

An example will make clear the construction and properties of this diagram. Fig. 410 shows five different configurations of one half of a Porter governor. The controlling force F for one ball for each of the five positions of the ball is determined by the construction shown in Fig. 411 for three different cases, namely, (1) when friction is neglected, (2) when friction is considered and the ball is falling, and (3) when friction is considered and the ball is rising. These values

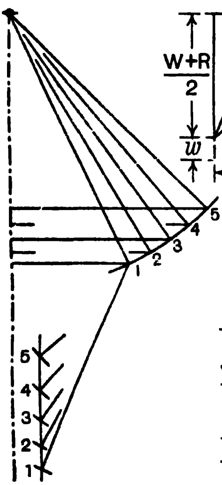


FIG. 410.

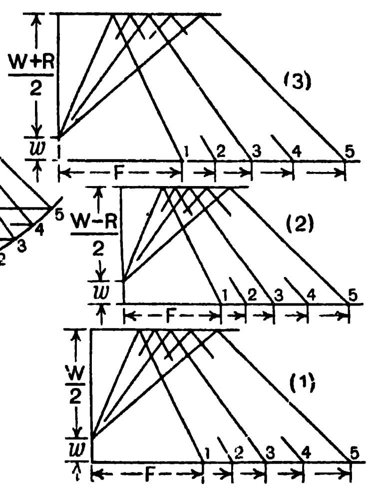


FIG. 411.

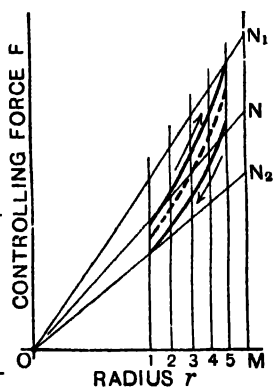


FIG. 412.

of F are plotted as ordinates and the corresponding values of the radius from Fig. 410 as abscissæ in Fig. 412. Of the three curves obtained in Fig. 412 the dotted one is the controlling force curve when friction is neglected. The lower and upper curves are the controlling force curves for the ball descending and ascending respectively when friction is considered.

The area of the diagram bounded by a controlling force curve, its extreme ordinates, and the base line, represents the work done by or against the controlling force while the ball moves over its horizontal range, and according to one definition of the power of a governor this area represents the power of the governor for one ball. The area of the diagram for one ball will represent the whole power of the governor if the ordinates of the curve are measured with a scale half that used for F in constructing the diagram. The power of the governor, in foot-pounds, is the mean height of the diagram (friction

neglected) in *pounds* on the force scale multiplied by the base in *feet* measured on the radius scale. The area between the upper and lower curves represents the work done on friction while the balls move outwards and back again over their horizonal range.

The curves of controlling force give further information about the governor when speed lines are added. A speed line is determined as follows. Choose a convenient radius OM (Fig. 412) and calculate the controlling force $\overline{MN}$ for the speed N from the formula $\overline{MN} = \frac{wr}{g}\left(\frac{\pi}{30}\right)^2 N^2$, and mark off $\overline{MN}$ on the force scale. If the speed N is kept constant while the radius is varied, the controlling force will be proportional to the radius, and for any radius will be represented by the corresponding ordinate of the straight line $\overline{ON}$. Conversely, if any straight line $\overline{ON}$ be drawn and considered as a speed line, the speed which it represents is calculated from the formula $N = \frac{30}{\pi}\sqrt{\frac{\overline{MN}\cdot g}{wr}}$. $\overline{ON}_1$ and $\overline{ON}_2$ are the maximum and minimum speed lines for this governor when friction is considered.

The point at which a particular speed line cuts a curve of controlling force gives the radius at which the balls will run at that speed for that curve.

For stability the slope of the controlling force curve at the point where it cuts any speed line must be greater than the slope of that speed line, and the greater the difference of these two slopes the greater is the stability of the governor, and the smaller this difference is the greater is the sensitiveness of the governor.

For certain spring-load governors the controlling force curves are straight lines whose equations are of the form $F = ar - b$, where a is a constant depending on the stiffness of the spring, and b is another constant depending on the initial load on the spring.

Fig. 413 illustrates two cases in which the controlling force lines are straight. In case (1) AB is the controlling force line when friction is neglected and DE and GH the controlling force lines when friction is considered. In this case the governor is stable.

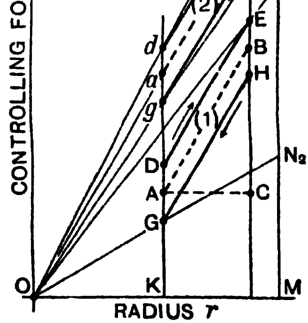


Fig. 413.

In case (2) *ab* is the controlling force line, neglecting friction, when the governor is isochronous. When a governor is isochronous its controlling force line where there is no friction passes when produced through the origin O.

It is important to notice that the slope of a straight controlling

force line is a measure of the stiffness of the spring of the governor. The stiffness of a spring is measured by the increase in the load divided by the deflection produced by that increase in load. Now the increase in the load is either equal to or proportional to the increase in the controlling force and the corresponding increase in the radius of the ball path is either equal to or proportional to the deflection of the spring. In the right-angled triangle ABC (Fig. 413) the increase in the controlling force is BC for an increase AC in the radius. Therefore the stiffness of the spring is represented by BC $\div$ AC. Another important point is that altering the initial load does not change the stiffness of the spring.

To make the governor isochronous the line AB must be raised, keeping its slope unaltered, until it reaches the position of *ab* which when produced passes through O. This requires that the initial controlling force KA shall be increased to K*a*.

If the controlling force lines *de* and *fg*, allowing for friction, be added in case (2) it is readily seen that the governor is no longer isochronous; also the governor is unstable when the balls are moving outwards but stable when the balls are moving inwards.

It may be left to the student to consider from a study of Fig. 413 the effects of the stiffness of the spring and the magnitude of its initial load on the stability, sensibility, and power of the governor.

Exercises XVII

1. Find the height, in inches, of a simple conical pendulum when the speed of rotation is 50 revolutions per minute. Determine also the decrease in height for an increase in speed of 4 per cent. and the increase in height for a decrease in speed of 4 per cent.

2. Three conical pendulums are shown in Figs. 414, 415, and 416. For the configurations shown the speed of revolution is the same for each pendulum. Calculate for each pendulum the percentage increase of speed which will raise the level of the ball 1·5 inches.

3. A direct loaded governor (Fig. 380, p. 320) has arms 11 inches long. Each ball weighs 4·5 lb. and the central load is 81 lb. Neglecting friction,

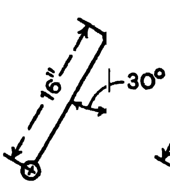


FIG. 414.

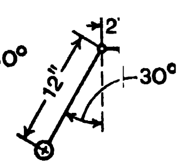


FIG. 415.

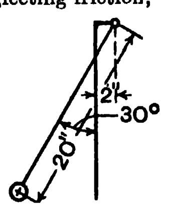


FIG. 416.

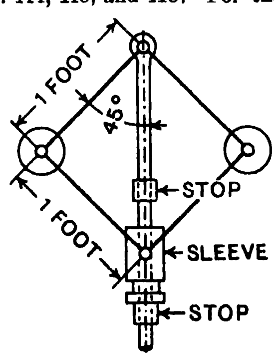


FIG. 417.

what is the speed in revolutions per minute when the arms are inclined at 30° to the axis of the spindle? Also what will be the upward displacement of the sleeve for an increase of 5 per cent. in the speed?

4. An equal-armed governor is shown in Fig. 417, with the sleeve resting against the bottom stop. Each ball weighs 4 pounds and the sleeve itself weighs 14 pounds. At what speed will the sleeve just begin to leave the bottom stop? [B.E.]

5. The sketch, Fig. 418, shows the leading dimensions of a loaded governor of the Porter type. Each ball weighs 4 pounds. Find the load W which must be applied to the sleeve so that the governor may revolve in the configuration shown at a speed of 200 revolutions per minute. [B.E.]

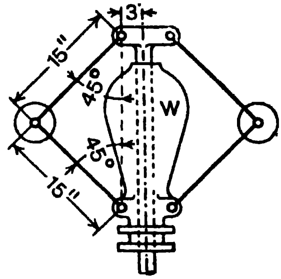


FIG. 418.

6. In a direct loaded governor of the type shown in Fig. 380, p. 320, the arms are 10 inches long. Each ball weighs 3·5 lb. and the central load is 70 lb. For the lowest and highest positions of the sleeve the arms are inclined at 30° and 40° respectively to the vertical. The friction of the governor and the mechanism connecting it to the valve is equivalent to a force of 4 lb. at the sleeve.

Find: (*a*) the travel of the sleeve; (*b*) the minimum ascending speed; and (*c*) the maximum descending speed; also, (*d*) the speed when the sleeve is at the middle of its travel and friction is neglected. Speeds to be in revolutions per minute.

7. In a governor of the type shown in Fig. 384, p. 322, $a = 12$ inches, $l = 20$ inches, and the suspension link is 12 inches long. The centre of the top joint of the arm and the centre of the bottom joint of the link are each 2 inches from the central axis of the governor. $W = 10$ lb. and $w = 12$ lb. The friction is equivalent to a force $R = 5$ lb. at the sleeve. For the lowest position of the sleeve the centre line of the ball arm is inclined at 21° to the vertical. The travel of the sleeve is 1·6 inches.

Determine the speed of the governor, in revolutions per minute, (*a*) when the sleeve begins to rise from its lowest position; (*b*) when the sleeve descending reaches its lowest position; (*c*) when the sleeve ascending reaches its highest position; (*d*) when the sleeve starts to descend from its highest position.

8. A Porter governor has arms and links each 15 inches long. The centres of the top joints of the arms are 3 inches apart and the centres of the bottom joints of the links are 4 inches apart. Each ball weighs w lb. and the central load is W lb. When the arms are inclined at 30° to the vertical and friction is neglected, find—(*a*) the speed N in revolutions per minute when $w = 10$ and $W = 80$; (*b*) W when $N = 150$ and $w = 10$; (*c*) w when $N = 144$ and $W = 95$.

9. Show for a loaded governor of the Porter type that when the central load rises and falls twice as fast as the balls

$$S = \sqrt{1+\lambda} - \sqrt{1-\lambda}$$

where S = the difference between the highest and lowest speeds in any position due to friction divided by the mean speed in that position, and λ = equivalent frictional resistance at the load divided by the weight of a ball plus the load. [U.L.]

10. Construct the speed curves described in Art. 229, p. 322, and illustrated by Fig. 385, for a Porter governor of which the particulars are as follows:—Arms and links 12 inches long. Distance between centres of top joints of arms and between centres of bottom joints of links, 3 inches. Weight of each ball, 6 lb. Central load, 60 lb. Friction equivalent to a force of 5 lb. at sleeve. For lowest position of sleeve the arms are inclined at 28° to the vertical. Travel of sleeve 2 inches.

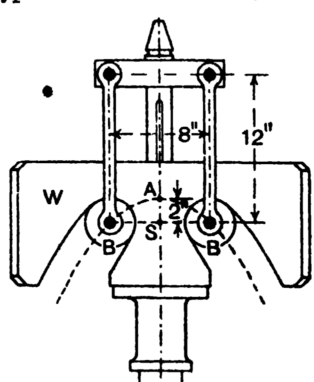


FIG. 419.

11. Galloway's parabolic governor is shown in Fig. 419. The balls B are in the form of cylinders and are suspended by links from a crosspiece attached to the central spindle as shown. While the centres of the balls move in arcs of circles their paths in relation to the central load W lie on a parabola, shown dotted. S is the focus and A the vertex of the parabola. The central load W weighs 350 lb. and each ball weighs 28 lb.

Determine the speed of this governor, in revolutions per minute, for the position shown and for lifts of 2, 4, and 6 inches.

[Note that the forces acting on one ball are: (1) The tension in the link, the vertical component of this being equal to half the central load plus the weight of the ball. (2) The pressure of the central load on the ball, the line of action of this being normal to the parabola. (3) The weight of the ball. (4) The centrifugal force of the ball.]

12. The sketch, Fig. 420, shows a loaded governor with crossed arms. The central load is 130 lb. and each ball weighs 15 lb. For the lowest position of the sleeve the angle θ is 38° and for the highest 46°. Find the lift of the sleeve. Compute the speeds of the governor, in revolutions per minute, for these two positions of the sleeve and also for the position for which θ is 42°, friction being neglected.

Also, assuming that the friction is equivalent to a force of 10 lb. at the sleeve, what are the extreme speeds for the lowest position of the sleeve?

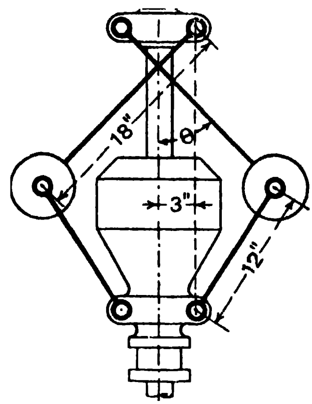


FIG. 420.

13. The arms and links of a governor form a parallelogram ABCD. AC is the axis of the governor and the sleeve is at C. The revolving balls at B and D each weighs 10 lb. What downward force Q must be applied to the sleeve, which weighs 5 lb., to prevent it from rising when the speed is increased by 2 per cent.? Also what will be the value of Q for the same increase in speed if there is an additional central load of 85 lb.? If the force Q is 5 lb. find the necessary percentage increase in speed to prevent the sleeve from descending in the case of the governor without the additional central load and also for the governor with the additional central load.

14. Find, for the position shown in Fig. 421, the speed at which the governor runs, neglecting friction and the weight of the arms. Explain why the governor is more sensitive than if the balls were placed at the junction of the links. The weight of each ball is 20 lb. and the load is 80 lb. [U.L.]

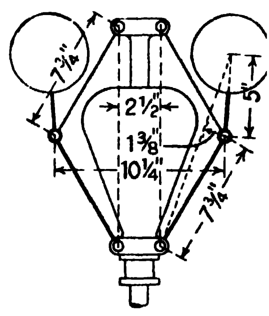


FIG. 421.

15. A Porter governor has arms and links each 11 inches long and the centres of the top and bottom joints are on the axis of the governor. Each ball weighs 5 lb. and the central load is 70 lb. Each arm and each link weighs 3 lb. Neglecting friction, find the speed of this governor, in revolutions per minute, when the arms are inclined at 30° to the vertical, (*a*) neglecting the masses of the arms and links, and (*b*) taking the masses of the arms and links into account.

16. In a spring governor of the Hartnell type, Fig. 422, the weight of each ball is 12 pounds, and the lift of the sleeve 2·5 inches. The speed at which the governor begins to float is 250 revolutions per minute, and at this speed the radius of the ball path is 5·5 inches. The mean working speed of the governor is 18 times the range of speed when the effects of friction are neglected. Calculate the stiffness of the spring, *i.e.* the load per inch compression. [U.L.]

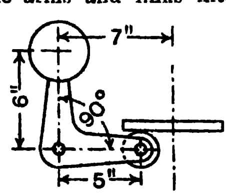


FIG. 422.

17. In a Hartnell governor the arms of the bell-crank levers are of equal length and are at right angles to one another. Each ball weighs 4 lb. Calculate the stiffness of the spring if the governor is isochronous at a speed of 280 revolutions per minute.

18. In the pendulum governor illustrated (Fig. 423), additional control is

obtained by a spring, stretching from centre to centre of the balls. The position shown is that of the governor at rest. If the weight of each ball is 3 lb., and the initial tension on the spring is 20 lb., determine the speed at which the balls begin to move outwards. If the spring requires 10 lb. per inch of extension, calculate the speed of the governor necessary to raise the sleeve two inches from its initial position, when the sleeve rises twice as fast as the balls. [U.L.]

19. One arm of one of the bell-crank levers of the governor illustrated by Fig. 406, p. 334, is shown in detail with dimensions in Fig. 424. Determine approximately the weight of a cylindrical mass which, placed with its axis on the axis of the hole in the larger end of the arm, will have the same centrifugal effect as that of the whole arm when the axis of the arm is parallel to the axis of the spindle and 6 inches distant from it. Take the weight of a cubic inch of the steel of which the arm is made as 0·28 lb.

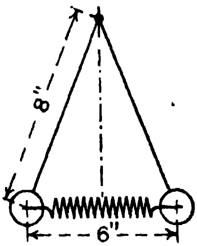


FIG. 423.

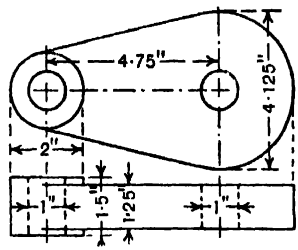


FIG. 424.

20. The governor, Fig. 406, p. 334, has a range of speed from 306 revolutions per minute when $r = 5·71$ inches to 314 revolutions per minute when $r = 6·29$ inches, when there is no tension in the adjustable supplementary spring. The centrifugal effect of the arms and weights (or balls), etc., is the same as that due to a weight of 20 lb. at the end of each ball arm. Find the stiffness of each of the two main springs and the equation giving the relation between the controlling force P and the radius r, P being in lb. and r in inches.

21. In a spring governor the effect of the spring, etc., in constraining the balls is found by observing the force which has to be applied radially at the balls to keep them in a given position. The observed force (neglecting friction) was 900 pounds when at 8 inches radius and 1500 pounds when at 12 inches radius, and the increase in the force was found to be proportional to the increase in the radius. Find the radius at which the balls run when the speed is 270 revolutions per minute, the weight of each ball being 60 pounds. Also, find by how much the initial tension in the spring must be increased in order to make the governor isochronous. At what speed would the governor then work? [U.L.

22. Construct the controlling force curve for the Porter governor of which particulars are given in Exercise 15, neglecting friction and the weight of the arms and links. Take the range of the radius of the ball path as from 5 inches to 7 inches. Add the speed line for 240 revolutions per minute. To show the influence of the central load add the controlling force curve when there is no central load. Use the following scales—Linear, 6 inches to a foot. Force, 10 lb. to an inch.

23. Taking the particulars given in Exercise 14 and Fig. 421 of a Proell governor, construct the controlling force curve for the range of the radius of the ball path corresponding to $\theta = 30°$ to $\theta = 40°$ where θ is the inclination of the links to the vertical. Linear scale 6 inches to one foot. Force scale 10 lb. to one inch.

24. Draw the Hartnell diagram for a spring-loaded governor of the type shown in Fig. 406, p. 334, for which the equation to the controlling force is $F = 80r - 170$, where F is the controlling force in lb. and r is the radius of the ball path in inches. The working range of the balls is from $r = 5·6$ to $r = 6·4$, and each ball weighs 22 lb. What is the power of this governor in foot-pounds? By how much must the initial load on each spring be increased to make the governor isochronous? Add to the diagram the speed line for 290 revolutions per minute and find the radius at which the balls will then be running.

CHAPTER XVIII

VALVES AND VALVE GEARS

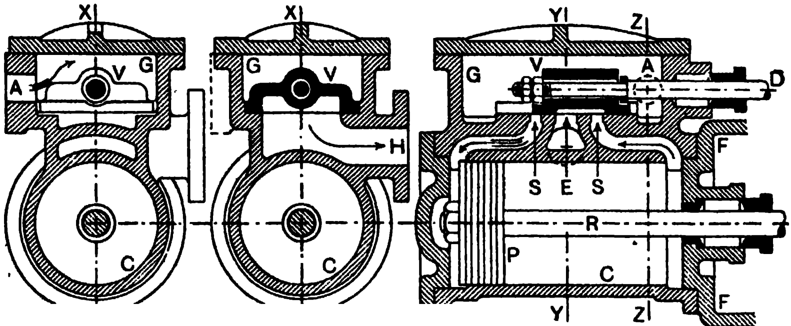
243. Functions of the Valves Considered in this Chapter.—The valves used in controlling the steam which drives the piston of a reciprocating engine have to perform four distinct operations on the steam used on one side of the piston, and the same operations have to be performed in the same order on the steam used on the other side for each cycle or each revolution of the crank shaft.

Stated briefly, the operations on the steam on one side of the piston are: (1) *Admission*, or opening of valve for the admission of the live steam to the cylinder. (2) *Cut off*, or stopping the admission of the steam previous to expansion. (3) *Release*, or opening of valve to allow the steam to escape from the cylinder to the atmosphere, or to the condenser, or to a larger cylinder. (4) *Compression*, or stopping the release of the steam from the cylinder previous to compression.

For a double acting piston there are eight valve operations per cycle or per revolution. All these eight operations may be performed by a single slide valve. Instead of a single slide valve two pairs of valves such as *Corliss valves* or *drop valves* may be used, one pair for each end of the cylinder, one valve at each end of the cylinder performing the operations (1) and (2), the other valves performing the operations (3) and (4). When a slide valve with what is called an *expansion valve* on the back of it is used the main valve performs all the operations except that of cut off for both ends of the cylinder, the function of the expansion valve being to control the cut off only.

244. The Simple Slide Valve.—A design of steam cylinder for use with a simple slide valve is shown in Fig. 425. C is the cylinder attached to the engine frame F. P is the piston and R the piston rod. On one side of the cylinder is the valve casing G containing the slide valve V which is connected to the valve spindle D. The valve may be described as a rectangular plate with a rectangular hole in it, this hole being walled in and covered over, the whole forming a single casting. The rectangular plate part of the valve fits against a flat seat formed on the bottom of the valve casing. This flat seat has three rectangular openings or ports in it, a central port E called the *exhaust port* and one S on each side of E called the *steam ports*. Each steam port is the entrance to a passage which leads to one end of the cylinder as shown. Steam enters the valve casing at A. In Fig. 425 the piston is at the left-hand end of the cylinder and the valve, which is moving to the right, has slightly uncovered the left-hand steam port allowing steam

to get on to the left-hand face of the piston which will begin to travel to the right. The hollow part of the valve being over the exhaust port and also over the right-hand steam port, the steam in the cylinder to the right of the piston is escaping through the exhaust port and thence through the exhaust pipe which is attached at H.



SECTION AT ZZ SECTION AT YY SECTION AT XX

FIG. 425.—Steam cylinder and slide valve.

Full consideration will be given later to the design and operation of the slide valve; meantime a few definitions relating to the slide valve may be given here.

A longitudinal section through a slide valve and cylinder ports is given in Fig. 426. The valve is shown in its middle position. The middle position of the valve is not necessarily that in which it is symmetrical with reference to the cylinder ports. This is generally the case in a horizontal engine but not so in a vertical engine. The middle position of the valve is that which it occupies when it is at the middle of its travel.

It will be seen that the valve more than covers the two steam ports. The amount *o* by which the valve overlaps a steam port on the outside or steam admission side is called the *outside lap* or *steam lap* of the valve for that port, the valve being in its middle position. The amount *i* by which the valve overlaps a steam port on the inside or exhaust side is called the *inside lap* or *exhaust lap* of the valve for that port, the valve being in its middle position. These definitions also apply to piston valves to be described presently, but since with these valves the steam admission is often on the inside and the exhaust on the outside of the valve the terms *steam lap* and *exhaust lap* are better general terms than *outside lap* and *inside lap*.

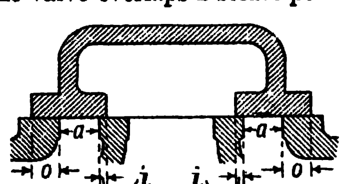


FIG. 426.

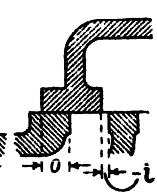


FIG. 427.

The inside or exhaust lap is frequently negative as shown in Fig. 427 where the valve does not entirely cover the steam port on the exhaust side when the valve is in its middle position.

When the engine piston is at one end of the cylinder and just about to begin a stroke, the steam port for that end of the cylinder is generally open by a slight amount e which is called the *lead* of the valve.

245. The Trick Valve.—Trick in Esslingen, Germany, in 1857 and Allan in England between 1858 and 1860 designed a modification of the simple slide valve which reduces the travel of the valve for the same port opening and therefore reduces the work on friction in driving it. The modification is simple and ingenious and is shown in Fig. 428.

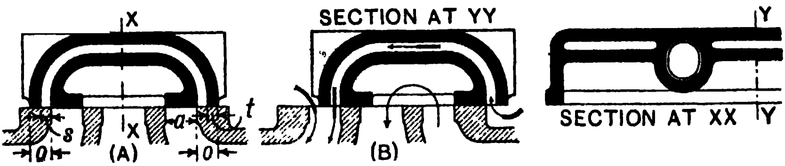


FIG 428.—The Trick valve.

The valve works on a raised seat and has an internal passage whose ports are in the valve face near the outer edges of that face. At (A) the valve is shown in its middle position and o is the outside lap. The valve shown has no inside lap. At (B) the valve is shown moved to its extreme right-hand position. The valve and the ports are so designed that as soon as the valve begins to uncover one steam port and steam enters it, as in the ordinary slide valve, steam also enters the same port from the internal passage, the steam entering this passage under the overhanging valve face at the other end.

The half travel of the valve is $o + s$, the effective port opening for steam is $2s$, and for exhaust $2s + t = a$.

246. Double Ported Slide Valve.—The principle of double and multiple ported slide valves is the same as that of the familiar sliding gridiron ventilator in which a comparatively small sliding movement gives a large area of opening. The port opening with a double ported slide valve is double that with a single ported valve for the same valve

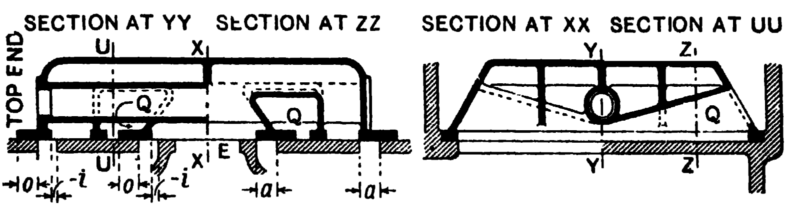


FIG. 429. -Double ported slide valve.

movement. Stated in another way, the sliding movement of a double ported valve is only half that of a single ported valve for the same port opening. This does not mean that the whole travel of the double ported valve is necessarily exactly half that of the single ported valve for the same total port opening but it is generally nearly if not exactly half.

A large double ported slide valve for a vertical marine engine is shown in Fig. 429, the valve being in its middle position. In con-

sequence of the weight of the piston, crosshead, and other reciprocating parts, it is usual to arrange the valve and its gear so that the work done by the steam below the piston is greater than that done above it. This explains why the valve has a greater outside lap at the top than at the bottom end; also why, while there is no inside lap at the bottom, the inside lap at the top end is negative.

To understand the action of the double ported slide valve it should be noted that at all times there is live steam all round the outside of the valve and also in the two passages Q which pass through the valve from one side to the other, while the remaining parts of the interior of the valve surrounding the tubular part through which the valve spindle passes contain exhaust steam.

By making a tracing of the ports and sliding this tracing into different positions under the longitudinal section of the valve the student should have no difficulty in following the action of the valve. It will be observed that there is only one exhaust port E, as in the single ported valve.

247. Balance Piston for Heavy Vertical Slide Valves.—The great weight of large slide valves, such as are found in vertical reciprocating marine engines, together with the weight of the valve gears, would put a great load on the eccentrics, and to avoid or reduce this load it is a common practice to introduce a *balance piston* in the manner shown in Fig. 430. T, the tail rod of the valve spindle, carries a piston P which works in the cylinder C. The piston, which is fitted with Ramsbottom packing rings is subjected to the valve chest pressure on its underside, and the top of the cylinder C being connected to the exhaust pipe of the main cylinder or to the condenser there is a resultant upward force on the balance piston which can be arranged to balance as much of the weight of the slide valve and its gear as may be desired by making the balance piston of suitable diameter.

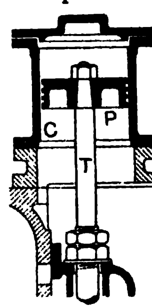


FIG. 430.

248. Pressure Relief Arrangements for Slide Valves.—The resultant force pressing a slide valve against its seat is the difference between the total pressure on the back of the valve and the total pressure on the exposed part of its face and interior. This resultant force varies with the position of the valve in relation to the steam and exhaust ports, but for any position in ordinary working it is much greater than is necessary to keep the valve against its seat. The consequence is that with an ordinary slide valve a considerable amount of work has to be done in overcoming the friction of the valve on its seat.

To reduce the resultant force pressing the valve on its seat, and therefore to reduce the work required to drive it, numerous arrangements have been devised. One feature of all these arrangements is that a portion of the area of the back of the valve is isolated from the steam chest pressure and subjected to a much lower pressure.

Fig. 431 shows a pressure relief arrangement for a locomotive slide valve. Four deep grooves of rectangular section are made in the back of the valve and into these are fitted four deep metal strips R which project slightly beyond the valve and are pressed outward by springs

against a smooth plate P cast in one with the valve chest cover C. The springs for forcing the strips R outwards may be waved steel strips placed in the grooves behind the packing strips, or they may be helical springs placed in holes at intervals as shown at S.

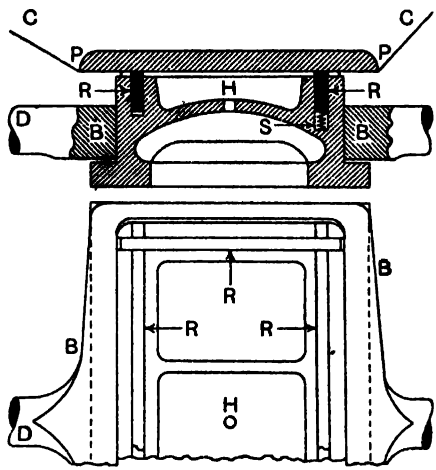


FIG. 431.

The packing strips enclose a rectangular area on the back of the valve which is in communication with the exhaust by means of the hole H, and this area is therefore only exposed to the exhaust pressure.

The packing strips used in the way described originated in the United States and are known as *Richardson strips.*

Fig. 431 also shows a very common way of connecting the slide valve spindle to the valve in locomotive practice. The spindle D has a rectangular frame B forged on to it and this fits over the valve as shown. The rectangular frame B is called a *buckle* or a *bridle.*

A relief ring used on the back of a double ported slide valve in marine practice is shown in Fig. 432. V, the back of the valve, has cast on it a shallow cylinder L which is bored and fitted with the relief ring R provided with Ramsbottom packing rings. The facing F on the back of the relief ring fits against a planed surface on the inside of the valve chest cover C. The relief ring is pressed against the valve chest cover by helical springs S. The space inside the relief ring is placed in communication with the condenser through the hole H.

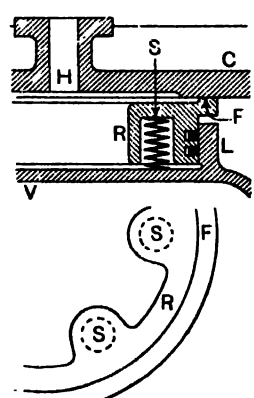


FIG. 432.

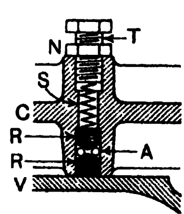


FIG. 433.

Another form of relief ring is shown in Fig. 433. V is the back of the valve. Two solid metal rings R fit into an annular recess cut in a projection on the inside of the valve chest cover C. Between the rings R two rings A of asbestos packing are placed and this combination of rings is pressed against the back of the

valve by helical springs S placed in holes at intervals above the rings. The force exerted by each spring is adjusted by a screw T which is secured in position by the lock nut N. The space within the relief ring communicates with the condenser as in Fig. 432.

249. Piston Valves.—Piston valves have now largely superseded the ordinary slide valve in reciprocating steam engines.

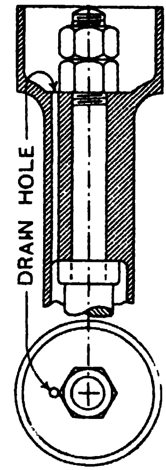


FIG. 434.

The simplest form of piston valve is the solid type, an example of which will be seen in the Allen engine illustrated by Fig. 223, p. 237. The upper part of the H.P. piston valve of this engine is shown to a larger scale in Fig. 434. This type works with less friction than any other and if the valve and the liner or casing in which it works are made truly cylindrical with a minimum difference in their diameters the leakage of steam past the valve should be very small, especially at high engine speeds.

One of the pistons of the piston valves of the G.E.R. 4–6–0 express passenger engine illustrated by Fig. 237, p. 213, is shown in detail in Fig. 435. This piston has a broad spring ring, the section of which is shown in solid black. The working diameter of this ring is 10 inches but it is first turned to an external diameter of 10 $\frac{9}{64}$ inches (about 10·14 in.) before it is cut. The ring is cut and machined to the form shown at (1), and when finished and closed to the working diameter the joint is as shown at (2). The facings on the ring and its internal flanges which bear on the body of the

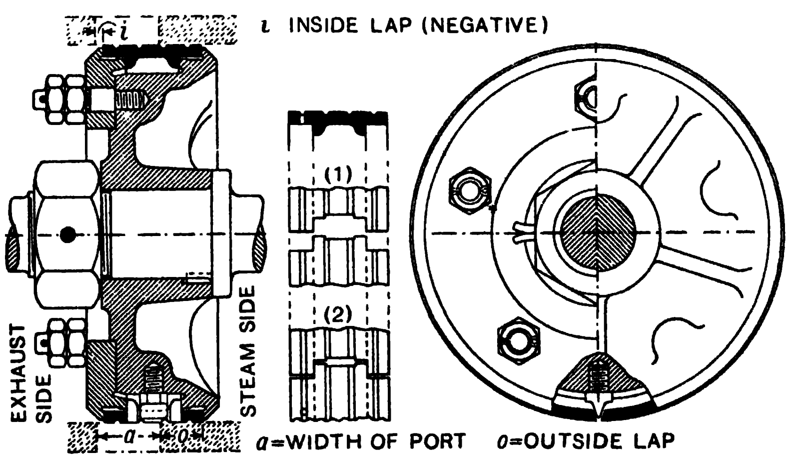


FIG. 435.—Piston of locomotive piston valve.

piston or on the junk ring are ground and finished before the ring is cut. The facings of the joint which come in contact when the ring is closed as at (2) are made an easy but steam-tight fit and their

planes are made to coincide with the planes of the facings of the internal flanges as shown.

When the piston is in position the split in the ring is at the bottom and at this part of the liner in which the piston works there is a broad bridge bar across the port. A stud screwed into the body of the piston and having a wedge-shaped head projecting into the split in the ring, as shown, keeps the ring in position.

Oil for the lubrication of the pistons is delivered under pressure through pipes which are connected to the valve casing at L (Fig. 237, p 243). Lubrication is facilitated by four narrow circumferential grooves on the piston ring. In each of these grooves three radial holes, 1–8th inch diameter, are drilled through the ring, the twelve holes being equally spaced round the ring.

During the admission of steam to the cylinder, especially at the beginning and end of the admission, the high steam pressure acting on the outside of the packing ring will tend to make it collapse and allow live steam to rush straight through to the exhaust side of the piston. If the ring is stiff enough to prevent any reduction in its diameter due to this external pressure then at other times the ring will exert too great a pressure on the liner and cause increased friction. The use of the small holes through the ring is to admit live steam to the annular space inside the ring and so balance the steam pressure on the outside. The pressure which the ring exerts on the liner is then due to the elasticity of the ring which is prearranged.

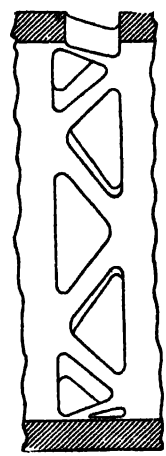


FIG. 436.

The steam port in the liner or bush in which the piston works has bridge bars across it at intervals solid with the liner, and these are preferably inclined as shown in Fig. 436, which represents the steam port in the liner for the piston shown in Fig. 435.

In the valve just described the live steam enters between the two pistons; this is called "inside admission." With inside admission the live steam is better protected than with outside admission.

A form of piston valve very common in American locomotive practice is shown in Fig. 437. Each piston consists of a "spider" S, a "bull-ring" B and two split packings rings P. The pistons are separated by a hollow body H. The pistons and the body are bound together by means of the valve spindle having a collar C and nuts N. T is the tail rod. Angular movement of the spider on the valve spindle is prevented by a key K and angular movement of the bull-ring on the spider is prevented by a key at XX on the vertical diameter. The split in the packing rings is at ZZ at the bottom of the piston and rotation of these rings is prevented by a pin passing through them at the split and through the bull-ring as shown. Cross sections at XX, YY, and ZZ to a larger scale are shown to the right in Fig. 437.

The hollow body H forms a communicating passage with the ends of the valve chest and with inside admission this passage contains exhaust steam but with outside admission it contains live steam. An

objection to a hollow piston valve is that the live steam on one side of the hollow body loses heat to the exhaust steam on the other side. On the other hand, the hollow body gives a simple means of communication between the two ends of the valve chest.

The great advantage which a properly constructed piston valve has over an ordinary slide valve is the small frictional resistance which it

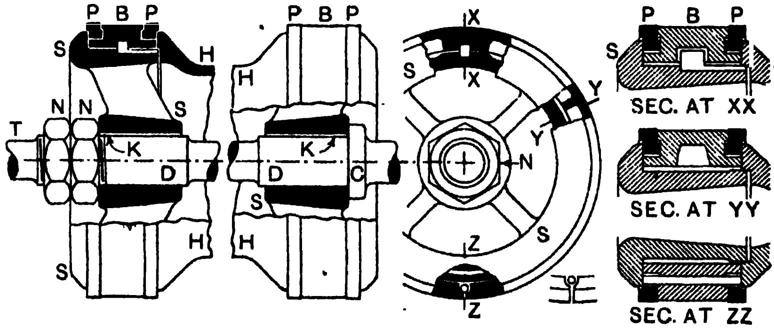


FIG. 437.—Locomotive piston valve.

offers. An ordinary slide valve will lift off its seat when, owing to abnormally high compression or the presence of water in the cylinder, the pressure on the face of the valve is greater than that on the back. The ordinary slide valve will therefore also act as a relief valve, an advantage not possessed by a piston valve as usually designed. It is therefore more important that the cylinder should be provided with relief valves when a piston valve is used instead of the ordinary slide valve.

The action of a piston valve in controlling the admission of the steam to the cylinder and its release is exactly the same as that of the ordinary slide valve, and the gear operating it may be the same. The outside and inside laps and the lead are also measured in the same way, and the Zeuner and other valve diagrams showing the operation of the valve are identical for the two types of valve.

250. Slide Valve and Piston Displacement.—A slide valve or piston having straight line reciprocating motion and connected to an eccentric or a crank by an eccentric rod or a connecting rod form one of the most common types of the mechanism known as the *slider-crank mechanism*, a mechanism which presents numerous interesting problems. The problem to be considered here is the determination of the distance moved or *displacement* of the slider (slide valve or piston) for a given displacement of the eccentric or crank.

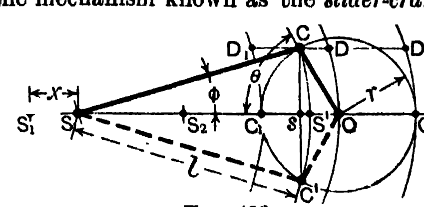


FIG. 438.

Referring to Fig. 438, O is the crank shaft, OC the crank, C the crank pin, and CS the connecting rod connecting the crank to the

slider S. The line of stroke S_1S_2 of the slider is assumed to pass, when produced, through O.

In considering the motion of the slider it is only necessary to fix attention on S where the connecting rod is jointed to the slider, because all points in the slider have exactly the same motion.

There are two positions of the mechanism where the crank and connecting rod are in line and these are called *dead centre positions*. In the case being considered the dead centre positions occur when C is at C_1 and at C_2 on the line of stroke, the corresponding positions of S being S_1 and S_2 for which S_1C_1 and S_2C_2 are equal to the length of the connecting rod.

The simple construction for finding S, the position of the slider for a given position OC of the crank, is to describe an arc of a circle whose centre is C and radius equal to the length of the connecting rod to cut the line of stroke at S. In the converse problem, given S, the position of the slider, to find the position of the crank, there are obviously two solutions, OC and OC′, excepting when the solution gives a dead centre position.

Since S_1S_2, the stroke of the slider, is equal to C_1C_2, the diameter of the crank pin circle, the position of the slider for a given position of the crank is conveniently shown on C_1C_2. Taking C_1C_2 to represent the stroke of the slider its position is represented by S′ where the arc of radius SC and centre S cuts C_1C_2. For $SS' = SC = S_1C_1$, therefore $S_1S = C_1S' = x$.

To avoid drawing the arc CS′C′ to find S′ for more than one position of C, the arc of radius equal to the length of the connecting rod which passes through O is drawn, then a horizontal CD to cut this arc at D gives $CD = S'O$ the distance of the slider from the middle of its stroke. If desired the distance of the slider from either end of its stroke for any position of C may be found at once after adding the arcs C_1D_1 and C_2D_2, struck from centres S_1 and S_2 respectively, then CD_1 is the distance of the slider from one end of its stroke, while CD_2 is its distance from the other end for the position OC of the crank.

It is evident that the greater the length of the connecting rod in relation to the radius of the crank the shorter will the distance sS' become, where s is the point where the chord CC′ cuts C_1C_2. CC′ is of course perpendicular to C_1C_2. In the case of a slide valve driven by an eccentric, which is a form of crank (see p. 266), the eccentric rod is generally so long compared with the radius or eccentricity of the eccentric that the distance sS' is small enough to be neglected. When S′ is taken as coinciding with s it is assumed that the slider has *harmonic motion.*

Treating the problem analytically it may be left as an exercise to the student to show that,

$$\begin{aligned} x &= r(1 - \cos\theta) + l(1 - \cos\phi) \\ &\doteq r(1 - \cos\theta) + \{l - \sqrt{l^2 - r^2\sin^2\theta}\} \\ &= r(1 - \cos\theta) + r\{n - \sqrt{n^2 - \sin^2\theta}\} \end{aligned}$$

where $n =$ the ratio of l to r.

Also for harmonic motion, $x = r(1 - \cos\theta)$.

251. Relative Positions of Eccentric and Crank.—The action of the simple slide valve, and the method of driving it, may now be further considered. A steam cylinder with ports and passages for a simple slide valve is shown in Fig. 439. The valve V has no lap,

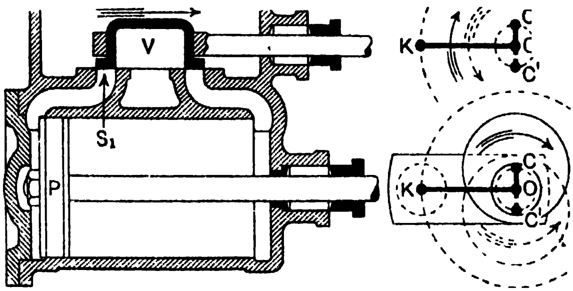


FIG. 439

outside or inside, and no lead. The piston P is shown at the left-hand end of its stroke and is just about to move to the right. In order that the piston may move to the right under steam pressure the left-hand steam port S_1 must be just on the point of opening and since the valve has neither lap nor lead it must be in its middle position as shown and must be moving to the right. The engine crank OK must evidently be in its inner dead centre position as shown and the eccentric, neglecting the obliquity of the eccentric rod, must be vertical, the axis of the cylinder being horizontal. There are two possible positions for the eccentric, one, OC, above the crank and the other, OC′, below, but these positions demand opposite directions of rotation for the crank shaft. In order that the valve may be moving to the right as required, the upper position OC of the eccentric requires clockwise rotation of the shaft while the lower position OC′ requires anti-clockwise rotation.

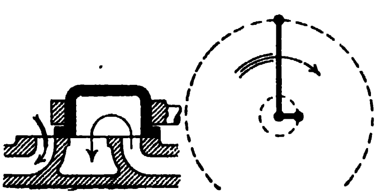


FIG. 440.

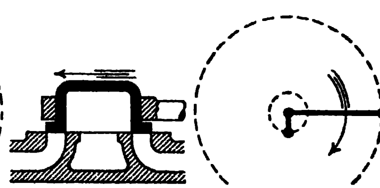


FIG. 441.

Taking clockwise rotation, the position of the valve and the positions of the crank and eccentric at the end of each quarter of a revolution from the initial dead centre position of the crank are shown in Figs. 440, 441, and 442. Without elaborating with further explanations, the student should be able to see clearly from a study of Figs. 439 to 442 that when a simple slide

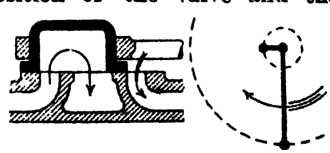


FIG. 442.

valve is used without lap or lead the eccentric must be fixed on the shaft 90° in advance of the crank, also that the steam admission on one side of the piston is continuous during the whole of a stroke, while the exhaust is continuous on the other side of the piston for the whole of the same stroke.

Now suppose that the valve is made with outside lap o as shown in Fig. 443 where the valve is supposed to be in its middle position. Imagine that the piston is at the left-hand end of its stroke as in Fig. 439 and assume that the crank shaft has clockwise rotation. The position of the piston requires that the left-hand steam port should be just on the point of opening, there being as yet no lead to the valve. Hence, without altering the position of the engine crank, the valve must be moved from its middle position a distance o to the right. This displacement of the valve may be attained, apparently, either by shifting the valve on its spindle or by shifting the eccentric on the shaft. A very little consideration will show that shifting the valve on its spindle, although it would enable the piston to start to the right under steam, would prevent the piston starting under steam from the other end of the stroke. The required displacement of the valve must therefore be effected by shifting the eccentric on the shaft as shown in

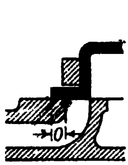


FIG. 443.

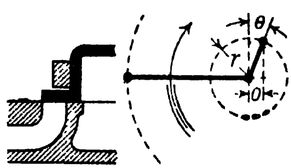


FIG. 444.

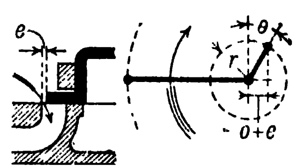


FIG. 445.

Fig. 444, the angle θ being such that $\sin \theta = o/r$, where r is the radius of the eccentric.

If the valve has lead as well as outside lap a further shifting of the eccentric is necessary as shown in Fig. 445, the angle θ being now such that $\sin \theta = (o + e)/r$.

The angle θ through which the eccentric must be advanced on the shaft to allow for the addition of outside lap or lead or both is called the *angle of advance of the eccentric.*

The effects of the addition of lap or lead or both to a valve on the admission, cut off, release, and compression of the steam in the cylinder are best studied by means of a *valve diagram,* forms of which will now be explained.

252. Zeuner's Slide Valve Diagram.—Fig. 446 shows part of a slide valve together with one steam port and part of the exhaust port. The circle X_1YXY_1, Fig. 447, represents the path of the centre of the sheave of the eccentric for driving this valve. These two illustrations are to the same scale. In what follows the obliquity of the eccentric rod is neglected. In Fig. 446 the valve is shown in three positions, (A) the middle position, (B) a position to the right, and (C) a position to the left of the middle position. The crank shaft is assumed to have clockwise motion as shown by the arrow in Fig. 447.

The horizontal diameter X_1OX is parallel to the line of travel of

the valve and is equal to that travel. YOY_1 is perpendicular to X_1OX. When the valve is in its middle position and moving towards the right the eccentric is in the position OY. Suppose the eccentric to move into the position OP. A perpendicular PR to OX determines OR the distance the valve has moved to the right from its middle position. The new position of the valve is shown at (B). On OP make OR′ equal to OR. If this construction is repeated for a number of positions of the eccentric as it turns from OY to OY_1 it will be found that the locus of R′ is a circle described on OX as diameter. That this is so is seen when the triangles OXR′ and OPR are compared. These triangles have two sides of the one equal to two sides of the other, each to each, and the angle at O between them common to both. Hence the angle OR′X is equal to the angle ORP which is a right angle. The locus of R′ is therefore a circle of which OX is a diameter.

In like manner repeating the construction while the eccentric turns from OY_1 to OY and the valve is moving on the left of its middle position, the locus of the point L′ is the circle described on OX_1

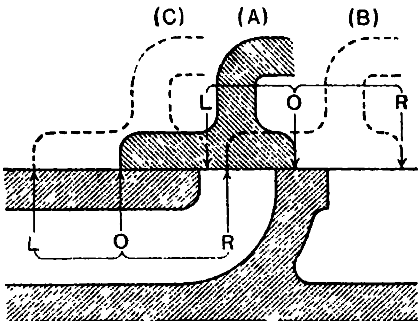


FIG. 446.

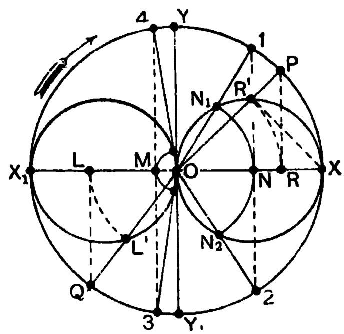


FIG. 447.

as diameter. By means of these circles on OX and OX_1, called *valve circles*, the position of the valve for any position of the eccentric can be found at once. For example, if OQ is the position of the eccentric then OL′ is the distance of the valve from its middle position, and since OL′ is on the circle on OX_1 the valve is to the left of its middle position.

In studying the operation of the valve the positions of its inner and outer edges have to be considered, and each of these has its own middle position O shown in Fig. 446.

If an arc of a circle N_1NN_2 be described, with centre O and radius equal to the outside lap, cutting the circle on OX at N_1 and N_2, then $ON_1 1$ and $ON_2 2$ are evidently the positions of the eccentric when the valve is just about to open for steam and when just about to close respectively. In like manner OM being equal to the inside lap, O3 and O4 are the positions of the eccentric when the valve is just about to open to exhaust and when it is just about to close respectively.

The diagram Fig. 447 gives the positions of the *eccentric* for the critical positions of the valve but a more useful diagram is one which

gives the positions of the *crank* for the critical positions of the valve. Now since the crank is $90^\circ + \theta$ behind the eccentric, where θ is the angle of advance of the eccentric, all that is necessary is that the diagram Fig. 447 be rotated backwards through an angle of $90^\circ + \theta$ and what were the positions of the eccentric will become the corresponding positions of the crank. This has been done in Fig. 448 where the diagram is drawn to a larger scale. This is the *Zeuner slide valve diagram.* The probable indicator diagram, neglecting the obliquity of the connecting rod, is shown beneath the valve diagram. In general the obliquity of the connecting rod cannot be neglected, but the true positions of the piston corresponding to the crank positions 1, 2, 3, and 4 may be found as explained in Art. 250, p. 349.

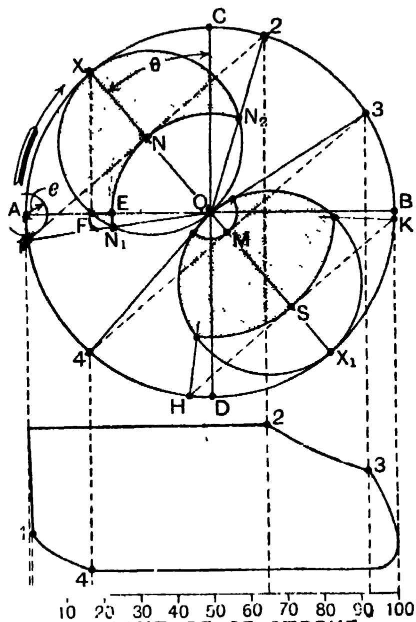


Fig. 448. Zeuner valve diagram.

The construction of the complete Zeuner valve diagram as shown in Fig. 448 may now be briefly summarized. AB is drawn to represent the travel of the valve and also, to a different scale, the stroke of the piston. On AB as diameter describe the travel circle ACBD of which CD is the diameter at right angles to AB. Imagine the steam cylinder to be to the left of the diagram and place an arrow against the travel circle to show the direction of motion of the shaft. Draw XOX_1 making the angle θ with CD, θ being measured from CD in the direction opposite to that of the arrow. The angle θ may be constructed by making OE equal to the outside lap o and EF equal to the lead e and drawing FX at right angles to AB. On OX and OX_1 describe the valve circles. Draw the outside and inside lap arcs. The positions 1, 2, 3, and 4 of the crank for admission, cut-off, release, and compression respectively may now be determined. Note that a circle with centre A and radius equal to the lead will touch the straight line 1N2. Make MS equal to the width of the steam port, and draw an arc through S with centre O. Lines through O and the points where the arc through S cuts the valve circle determine the angle KOH through which the crank turns while the steam port is full open to exhaust. A similar construction on the other valve circle would determine the angle through which the crank turns while the steam port is full open to steam, but in the case illustrated the steam port is not full opened to steam. The maximum opening of the port to steam is NX.

The diagram shown in Fig. 448 is for the steam on one side of the piston only or for one half of the valve. To draw the diagram for the other half of the valve the same valve circles are used but the two circles and the lines associated with them change places. Note that although the laps for one half of the valve may not be the same as those for the other, θ is the same. Hence $e + o$ must be the same for both halves of the valve.

For the sake of clearness the valve diagrams for the two ends of the cylinder may be drawn separately.

253. Reuleaux's Slide Valve Diagram.—Referring back to Fig. 447 it was shown that for the critical positions of the valve, namely, admission, cut-off, release, and compression, the corresponding positions of the eccentric are, O1, O2, O3, and O4, respectively, the points 1, 2, 3, and 4 being obtained by drawing lines 1 2, and 3 4 at right angles to X_1X or parallel to YY_1 and at distances from YY_1 equal to the outside and inside laps respectively. To obtain the corresponding positions of the crank it is only necessary to rotate the diagram in the opposite direction to that of the rotation of the shaft through an angle equal to $90° + \theta$, where θ is the angle of advance of the eccentric. This gives the *Reuleaux valve diagram* shown in Fig. 449. If this be compared with the Zeuner diagram shown in Fig. 448 it will be seen that the two diagrams are very much alike, and it will be found that problems which may be solved by the one may be solved by the other equally well.

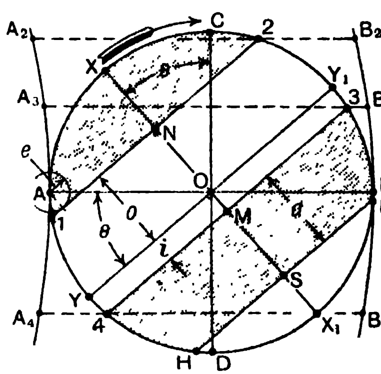


FIG. 449.—Reuleaux valve diagram.

The steps in the construction of the Reuleaux diagram, when all the particulars of the valve are known, are as follows. Draw AB to represent the stroke of the piston and also the travel of the valve. Describe the circle ACBD on AB as diameter. Draw the diameter CD at right angles to AB. Draw XOX_1 making the angle COX equal to θ the angle of advance of the eccentric, the angle COX being measured from CO in the direction opposite to that of the arrow which shows the direction of rotation of the shaft. Make ON equal to o the outside lap, OM equal to i the inside lap, and MS equal to a the width of the steam port. Draw 1N2, 4M3, and HSK at right angles to XX_1. Then the points 1, 2, 3, 4, H, and K are the same as those similarly designated in the Zeuner diagram, Fig. 448.

If the obliquity of the connecting rod be neglected the positions of the piston corresponding to the crank positions 1, 2, 3, and 4 are obtained by dropping perpendiculars from these points on to AB. The true positions of the piston are quickly found by using the arcs A_2AA_4 and B_2BB_4 as explained in Art. 250, p. 349.

254. Bilgram's Slide Valve Diagram.—Referring to Fig. 450, AB

represents the valve travel and also the piston stroke, and the circle described on AB as diameter is the travel circle Suppose that the cylinder is to the left of the diagram and that the shaft is rotating in the direction of the arrow. At (*a*), Fig. 450, if ON to the right of O is equal to the outside lap of the valve and PNQ is at right angles to AB, OP is the position of the eccentric at admission and OQ is its position at cut-off. O1 and O2, the corresponding positions of the crank are obtained by measuring backwards from OP and OQ respectively the angle $90° + \theta$, where θ is the angle of advance of the eccentric. If the angle between 1O and 2O produced be bisected by OC it will be found that the angle AOC is equal to θ. Perpendiculars CD and CD′ drawn to 1O and 2O produced are equal and a circle with centre C and radius equal to CD or CD′ will touch the lines 1O and 2O produced. By comparing the triangles CDO and ONP it may be proved that CD is equal to ON. The radius of the circle whose centre is C and which touches the lines 1O and 2O produced is therefore equal to *o* the outside lap.

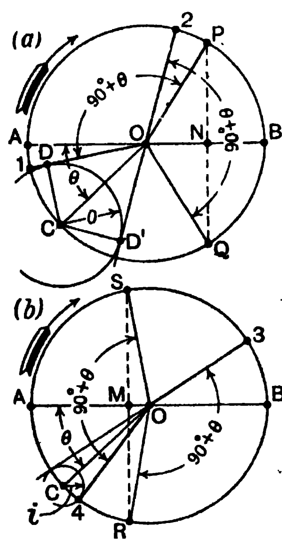


FIG. 450.

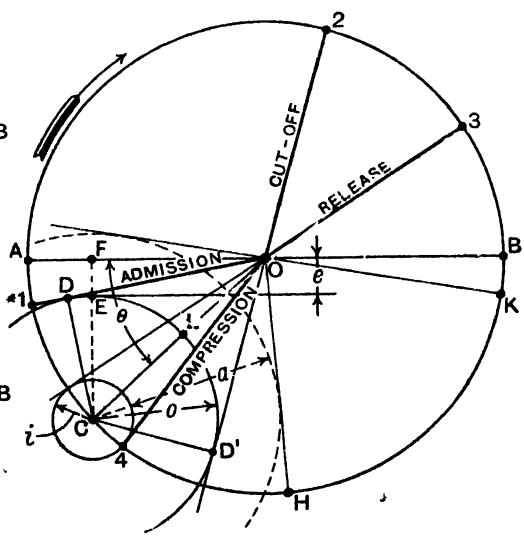


FIG. 451.—Bilgram valve diagram.

In like manner if at (*b*), Fig 450, OM to the left of O is equal to the inside lap of the valve and RMS is at right angles to AB, OR is the position of the eccentric at release and OS is its position at compression. O3 and O4 the corresponding positions of the crank are obtained by measuring backwards from OR and OS respectively the angle $90° + \theta$. If the angle between 3O produced and 4O be bisected by OC it will be found, as before, that the angle AOC is equal to θ. Also the circle whose centre is C and which touches the lines 3O produced and 4O has a radius equal to *i* the inside lap.

The foregoing are particular cases of a general rule, namely, that for any position of the crank the perpendicular on it from the point C

gives the distance of the valve from its middle position for that position of the crank.

These results lead to the *Bilgram slide valve diagram* shown in Fig. 451. Assuming that all the particulars of the valve are known and that the crank positions for admission, cut-off, release, and compression are required, the steps in the construction of the Bilgram diagram are as follows. Make AOB to represent the valve travel and also the piston stroke. On AB draw the travel circle. Draw OC making the angle AOC equal to θ. With centre C and radii equal to o and i, the outside and inside laps, describe the lap circles. Tangents to these circles through O determine the required crank positions as shown.

If a circle be drawn concentric with the lap circles and at a distance from the inside lap circle equal to a the width of the steam port, then tangents to this circle through O determine the angle KOH through which the crank turns while the steam port is full open to exhaust.

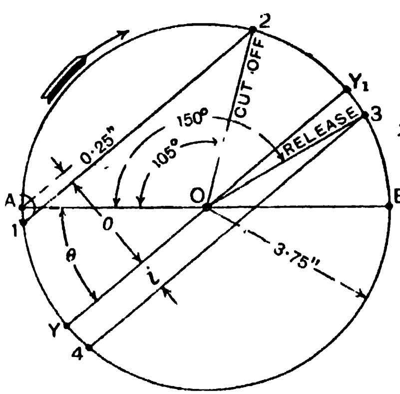


Fig. 452.

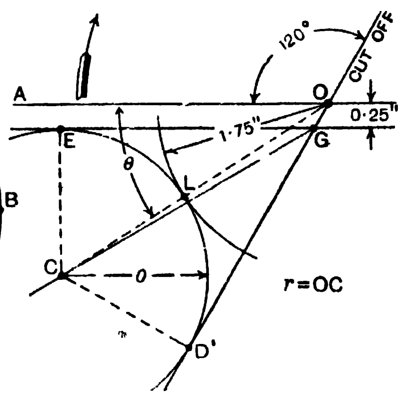


Fig. 453.

OL which is less than a is the maximum opening of the steam port to steam.

It may be noted that if CF be drawn at right angles to AB cutting the outside lap circle at E, then EF is equal to the lead of the valve.

255. Slide Valve Problems.—To further illustrate the use of the slide valve diagrams which have been described, a few slide valve problems will now be worked out graphically. To draw a slide valve diagram it is not necessary to know all its elements, but if a sufficient number of the elements be given the diagram may be drawn and the remaining elements determined.

As to the type of diagram to use for any particular problem, the Reuleaux diagram is in general the most convenient, if the travel of the valve is known, although the Zeuner or Bilgram diagrams may also be used. If, however, the travel of the valve is unknown the Bilgram diagram is the one to use for a direct solution.

Problem I. (Fig. 452).—Given: Position of crank at cut-off, 105° and at release, 150° from inner dead centre position. Radius of eccentric, 3·75 inches. Lead, 0·25 inch.

Required: Outside and inside laps and the angle of advance.

Using the Reuleaux diagram, draw the travel circle AY_1BY and take the diameter AB as the line of travel. With centre A and radius equal to the given lead describe a circle. From 2 draw 21 to touch this lead circle. Draw Y_1OY and 34 parallel to 21. The perpendicular distances of Y_1Y from 21 and 34 are the required outside and inside laps o and i respectively, and the angle AOY is the angle of advance θ.

$$o = 2{\cdot}15 \text{ inches.} \quad i = 0{\cdot}64 \text{ inch.} \quad \theta = 39^\circ\ 54'.$$

PROBLEM II. (Fig. 453).—Given: Position of crank at cut-off, 120° from inner dead centre position. Lead, 0·25 inch. Maximum opening of port to steam, 1·75 inches.

Required: Outside lap, and radius and angle of advance of eccentric.

Using the Bilgram diagram, draw the line AO of indefinite length to represent the line of travel and draw EG parallel to AO and at a distance from it equal to the given lead. Draw the line representing the crank at cut-off and produce it through O indefinitely, cutting EG at G. With centre O and radius equal to the given maximum port opening describe an arc of a circle as shown. Draw GC bisecting the angle EGD′. The required lap circle touches the maximum port opening arc at L, the line EG at E, and the line GD′ at D′. The centre C of the lap circle lies on the line GC. There is no simple direct construction for finding C, but it may be found readily and with great accuracy by trial.

$$o = 1{\cdot}49 \text{ inches.} \quad r = OC = 3{\cdot}24 \text{ inches.} \quad \theta = 32^\circ\ 29'.$$

PROBLEM III. (Fig. 454).—Given: Travel of valve, 7·2 inches. Lead at both ends, 0·4 inch. Outside lap at both ends, 2 inches. Inside lap at both ends, 0·6 inch. Length of connecting rod equal to twice stroke of piston.

Required: Percentage of stroke of piston at cut-off, release, and compression, of steam on both sides of piston.

Using the Reuleaux diagram, draw the travel circle AY_1BY and take the diameter AB as the line of travel. With centre A and radius equal to the given lead describe a circle. With centre O and radius equal to the given outside lap describe another circle. The line 12 touching these two circles determines 2, the position of the crank pin at cut-off for the cover side of the piston. The line 2′1′ parallel to 12 and touching the outside lap circle determines 2′, the position of the crank pin at cut-off for the crank side of the piston.

With centre O and radius equal to the given inside lap describe a circle. Tangents to this inside lap circle parallel to 1 2 determine 3 and 3′ the positions of the crank pin at release for the cover and crank sides of the piston respectively, and also 4 and 4′ the positions of the crank pin at compression for the cover and crank sides of the piston respectively.

Arcs of circles having a radius equal to twice AB and having their centres in BA produced and passing through the above-mentioned crank pin positions determine the corresponding piston positions C, C′, R, R′, S and S′.

The required results are as follows ;—

Cover side of piston.

$$\text{Cut-off} = \frac{AC}{AB} \times 100 = 68{\cdot}1 \text{ per cent.}$$

$$\text{Release} = \frac{AR}{AB} \times 100 = 93{\cdot}9 \text{ per cent.}$$

$$\text{Compression} = \frac{BS}{BA} \times 100 = 77{\cdot}2 \text{ per cent.}$$

Crank side of piston.

$$\text{Cut-off} = \frac{BC'}{BA} \times 100 = 56{\cdot}5 \text{ per cent.}$$

$$\text{Release} = \frac{BR'}{BA} \times 100 = 90{\cdot}7 \text{ per cent.}$$

$$\text{Compression} = \frac{AS'}{AB} \times 100 = 84{\cdot}7 \text{ per cent.}$$

AOY is the angle of advance of the eccentric, YOY_1 being parallel to 1 2.

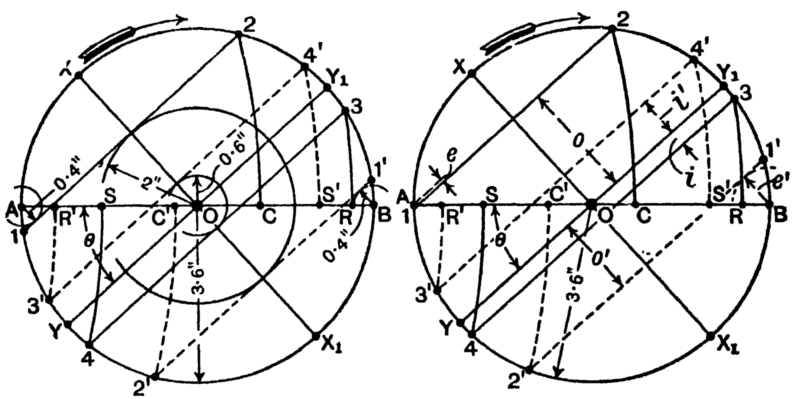


FIG. 454. FIG. 455.

PROBLEM IV. (Fig. 455).—Given: Travel of valve, angle of advance of eccentric, and length of connecting rod the same as in Prob. III. Cut-off on both sides of piston to be the mean of the cut-offs found in Prob. III. Release on both sides of piston to be the mean of the releases found in Prob. III.

Required: Outside and inside laps, and leads, for both ends of valve, also points of compression for both sides of piston.

The mean of the two cut-offs found in Prob. III. is

$$\frac{68{\cdot}1 + 56{\cdot}5}{2} = 62{\cdot}3 \text{ per cent.}$$

Make AC and BC′ each equal to 62·3 per cent. of AB.

The mean of the two releases found in Prob. III. is

$$\frac{93{\cdot}9 + 90{\cdot}7}{2} = 92{\cdot}3 \text{ per cent.}$$

Make AR and BR′ each equal to 92·3 per cent. of AB.

Allowing for the obliquity of the connecting rod the crank pin positions 2, 2′, 3, and 3′ corresponding to the piston positions C, C′, R, and R′ are next found.

YY_1 is drawn making the angle AOY the same as found in Prob. III. Lines through 2, 2′, 3, and 3′ parallel to YOY, determine the leads e, e', and laps o, o', i, and i'. Piston positions S and S′ at compression are found from the crank pin positions 4 and 4′.

The required results are as follows:—

Cover end.—Lead $e = 0·04$ inch. Outside lap $o = 2·36$ inches. Inside lap $i = 0·33$ inch. Compression $= \frac{BS}{BA} \times 100 = 80·6$ per cent.

Crank end.—Lead $e' = 0·76$ inch. Outside lap $o' = 1·64$ inches. Inside lap $i' = 0·8$ inch. Compression $= \frac{AS'}{AB} \times 100 = 83·1$ per cent.

256. Consequences of Early Cut-off with Slide Valve and Single Eccentric.—An examination of a valve diagram for a slide valve, driven directly by a single eccentric, when the cut-off is, say, at quarter stroke, shows that there are several disadvantages attending an early cut-off by this means. Referring to Fig. 456, which is the Reuleaux diagram for a slide valve arranged to cut-off at quarter stroke, neglecting the obliquity of the connecting rod, it is seen that the maximum port opening NX is very much reduced from what it would be with a cut-off at, say, three-quarter stroke with the same lead and valve travel. The consequence of this is a throttling of the steam as it enters the cylinder.

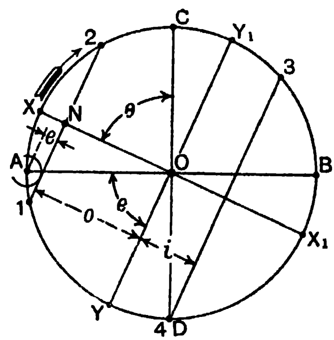


FIG. 456.

Another point to observe is that at cut-off the eccentric radius is inclined at a small angle to the line of valve travel and the velocity of the valve is therefore much less than its maximum velocity which occurs when the eccentric radius is at right angles to the line of valve travel, neglecting the obliquity of the eccentric rod. The result of this is an absence of sharpness in the cut-off and a further throttling of the steam.

It will also be seen that the necessarily large angle of advance of the eccentric necessitates a large inside lap in order to avoid a too early release and a consequence of this is a large amount of compression during the return stroke.

In practice the slide valve driven directly by a single eccentric is seldom designed to cut off earlier than half stroke.

257. The Meyer Expansion Valve—An early cut-off is very advantageously obtained by the use of a supplementary slide valve working on the back of another. The supplementary valve controls the cut-off only and is called the *expansion valve*, the other being called the *main valve*. The functions of the main valve are to control the admission.

release and compression. The two valves are driven by separate eccentrics. Of this type of valve combination the Meyer design has been largely used especially when arranged for variable expansion.

The Meyer arrangement for variable expansion is shown in Fig. 457. AB is the main valve, the central part CD being an ordinary slide valve. The extensions of the main valve beyond the central part provide two ports through which the steam must pass before entering the steam ports leading into the cylinder. The main valve has outside lap *o* and inside lap *i* and is designed with the eccentric which drives it to given definite admission, cut-off, release, and compression, the cut-off being a late one generally not less than about three-quarter stroke. These operations of the main valve are not subject to alteration.

The expansion valve on the back of the main valve consists of two main parts which are flat plates M stiffened by ribs and carrying nuts N engaging with square-threaded screws T and U on the expansion valve spindle Ss. Of the two screws T and U, one is right- and the other left-handed. The main and expansion valves may or may

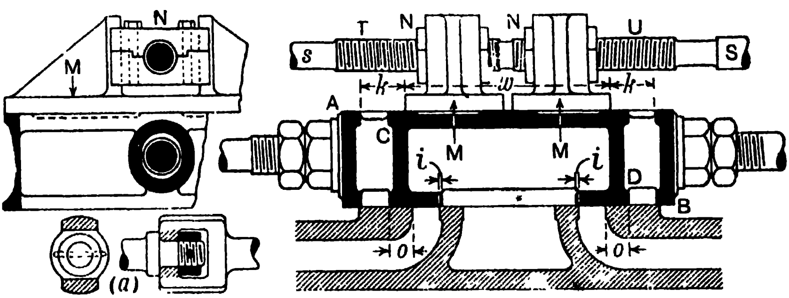


FIG. 457.—Meyer expansion valve.

not have the same travel but their eccentrics have different angles of advance. While the main eccentric may have an angle of advance of, say, 30°, the expansion eccentric will have an angle of advance in the neighbourhood of 90°.

Provided that the total width *w* over the expansion plates is sufficient the relative motion of the two valves will cause the expansion valve to cover alternately the ports in the main valve and therefore cut-off the steam to the cylinder although the main valve may be open to steam. The action of the expansion valve will be more fully understood when the valve diagrams discussed in the next Art. are considered, but for the present it may be stated that the point at which the steam is cut off by the expansion valve may be varied by altering the width *w*.

In Fig. 457 the main and expansion valves are shown each in its middle position although when working they are never simultaneously in that position. The valves being in the position shown in Fig. 457, the distance *k* between an outer edge of the expansion valve and the edge of the port in the main valve with which it coincides at cut-off by the expansion valve will for brevity be called the *gap* of the expansion

valve. As shown in Fig. 457 the gap is positive but it may be negative. The gap is negative when the expansion valve overlaps the ports in the main valve, both valves being in their middle positions.

At cut-off by the expansion valve it is evident that the difference between the distances of the valves from their middle positions must be equal to the gap. The gap may be varied by rotating the expansion valve spindle. For each particular cut-off by the expansion valve there is a particular value for the gap and it will be shown in the next Art. how this value may be obtained.

Fig. 458 shows an arrangement for altering the gap by hand, its operation being independent of whether the engine is running or not. An extension *s* of the expansion valve spindle passes through a stuffing-box to the outside of the valve chest and then through a sleeve E carried by a bracket K which is fixed to the outside of the valve chest. For about half its length the hole in the sleeve is square, and for part of its length the spindle *s* is also square and is a sliding fit in the square hole in the sleeve. A hand wheel H is keyed to the sleeve and is further secured by a nut G which is extended to form a cap to

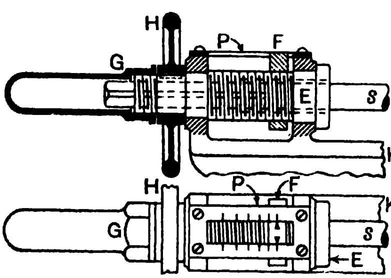


FIG. 458.

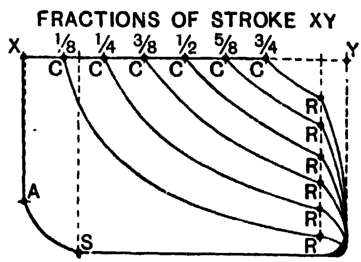


FIG. 459.

guard the end of the reciprocating spindle. This arrangement enables the valve spindle to be rotated without interfering with its freedom to reciprocate. To permit of the rotation of the valve spindle a swivel joint, such as is shown at (*a*) in Fig. 457, is provided at its other end.

The central part of the sleeve E is screwed externally and works in a nut F, which may slide but is prevented from rotating by a projection which fits into a groove in the plate P. The position of the nut F is shown on a scale engraved on the index plate P, and since this position corresponds to a particular gap of the expansion valve and also to a particular cut-off, it is this cut-of which is marked on the scale.

The changes in the form of the indicator diagram due to changes in the gap of the expansion valve are exhibited in Fig. 459. It will be noticed that A, R, and S, the points of admission, release, and compression, which are controlled by the main valve, occupy the same positions in the stroke for different positions of C, the point of cut-off which is controlled by the expansion valve.

258. Zeuner Diagram for Meyer Expansion Valve.—The design and operation of the Meyer expansion valve may be conveniently

studied by means of the Zeuner valve diagram. Referring to Fig. 460 the circles on OX and OX_1 as diameters are the valve circles for the main valve, θ being the angle of advance of the main eccentric. The circles on OV and OV_1 as diameters are the valve circles for the expansion valve, ϕ being the angle of advance of the expansion eccentric.

Let OP be any position of the crank cutting the expansion valve circle at L and the main valve circle at M, then for this position of the crank the distance of the main valve from its middle position is OM and the distance of the expansion valve from its middle position is OL. Therefore LM is the relative displacement of the expansion valve on the main valve, the relative displacement being measured from the position where both valves are central as in Fig. 457. Hence if cut-off by the expansion valve takes place when the crank is in the position OP, the gap k (Fig. 457) must be equal to LM.

To find the position of the crank at cut-off by the expansion valve for a given value of k is not quite so simple with the diagram as drawn in Fig. 460, but it may readily be found by trial.

By the addition of another circle to the diagram in Fig. 460 it is

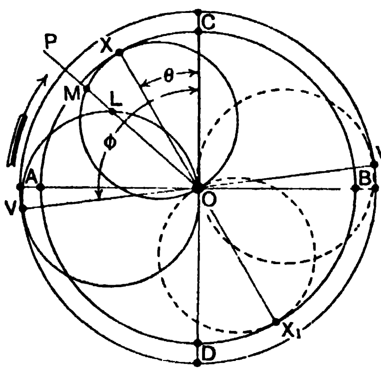


FIG. 460.

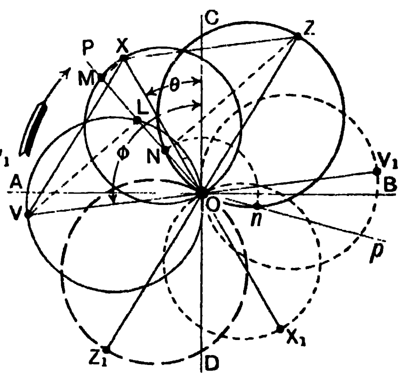


FIG. 461.

improved in that the crank position at cut-off by the expansion valve for a given value of k may be found directly. Referring now to Fig. 461 in which the valve circles in Fig. 460 are repeated, draw OZ parallel to VX, and XZ parallel to VO, thus forming a parallelogram OVXZ. On OZ as diameter describe a circle. Produce ZO to Z_1 and make OZ_1 equal to OZ. On OZ_1 as diameter describe a circle. These two additional circles are called *relative motion circles* and are the valve circles for the expansion valve if driven by an imaginary eccentric which gives to the expansion valve the same motion as before in relation to the main valve, but the main valve is now supposed to be at rest. Take any position OP of the crank cutting the various circles at L, M and N as shown. Join VL, XM, and ZN. The three lines VL, XM, and ZN are perpendicular to OP since angles in semicircles are right angles. LM is the projection of VX on OP and ON is the projection of OZ on OP, but OZ is equal and parallel to VX, therefore ON is equal to LM. The main and expansion valve circles may now be discarded and the relative motion circles

used to find the relative displacement of the expansion valve on the main valve for a given crank position or conversely.

The following obvious results should be noted. (1) If cut-off by the expansion valve takes place when the crank is in the position OP then k must be equal to ON. (2) If a circle be described with centre O and radius ON cutting the same relative motion circle again at n, then Onp is the position of the crank when the expansion valve is just about to uncover the port in the main valve. (3) If k is greater than OZ the expansion valve will not close the port in the main valve for any position of the crank. (4) When the crank position, for cut-off by the expansion valve, is tangential to the relative motion circles at O the value of k is nothing and for earlier positions of the crank for cut-off in the same stroke the values of k are negative.

The construction for finding the divisions of the scale on the index plate P in Fig. 458 is shown in Fig. 462. The circle described on OZ as diameter is the relative motion circle and AB represents the stroke of the piston. For clearness the main and expansion valve circles and the construction for finding OZ are omitted. For the case worked out in Fig. 462 the travel of the main valve is 3·2 inches, the travel of the expansion valve is 3·6 inches, $\theta = 30°$ and $\phi = 90°$. The obliquity of the connecting rod is neglected. The values of k are transferred to OA as shown and the numbers attached are the corresponding points of cut-off in eighths of the stroke of the piston. The actual values of k in inches are found from the scale at the bottom of the diagram, OT is the crank position when k is zero, and OP is a crank position corresponding to a negative value of k.

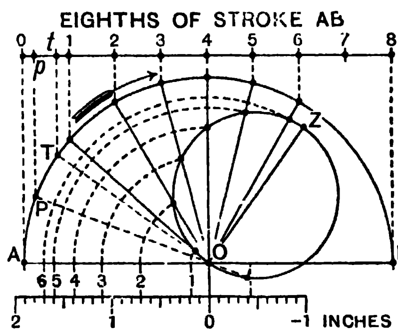


FIG. 462.

It is obvious that the latest effective cut-off by the expansion valve cannot be later than the cut-off by the main valve.

In order that the scale of gaps for the various cut-offs determined above may be the one to engrave on the index plate P, Fig. 458, it is necessary that the pitch of the screw on the sleeve E be the same as that of the screws on the expansion valve spindle. If it is desired to use on E a different pitch of screw all that is required is that the scale as determined above be altered in the ratio of the pitch of the screw on E to the pitch of the screws on the expansion valve spindle.

When the obliquity of the connecting rod is allowed for, it is of course found that the gaps of the expansion valve are different for the same points of cut-off on the two sides of the piston. There are several ways of approximately equalizing the gap scales for the two strokes. One way is to take the mean gap for each particular cut-off as the gap to be used in constructing the scale for the index plate. Another way is to construct the scale to suit one stroke and alter the position of the

expansion plate for the other stroke so that both plates have the correct gaps for one particular cut-off, say that at which the engine will most frequently run. The gaps of the two expansion plates will then have a constant difference. The constant difference may also be selected after an inspection of the various differences between the correct gaps for the respective cut-offs, that difference which seems to give the best average result being selected.

Care must be taken in designing a Meyer expansion valve that the width of the expansion plates is sufficient to prevent the ports in the main valve becoming uncovered, at the inside edges of the plates, after the earliest cut-off and before cut-off by the main valve. Let m denote the minimum width of plate to prevent readmission of steam before cut-off by main valve, K the relative motion of the expansion valve on the main valve at cut-off by the main valve, k_1 the value of the gap k for the earliest cut-off, and a the width of the ports in the main valve, then from Fig. 463 it is evident that $m = K - k_1 + a$.

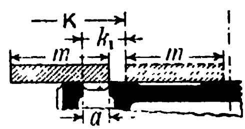


FIG. 463.

259. Movable Eccentric.—An eccentric sheave may be attached to the crank shaft in such a way that the position of its centre may be readily varied so as to alter the point of cut-off in the cylinder, and by carrying the displacement of the sheave far enough the engine may be made to run in the reverse direction. An arrangement of this kind, to be altered by hand when the engine is at rest, is shown in Fig. 464. The eccentric sheave C has two slots in it, one to clear the shaft S and the other to clear a bolt B which passes through a flange F forged on the shaft. C_1 and C_2 are the extreme positions of the centre of the sheave, C_1 being for full power forward and C_2 for full power backward. The thick line KO shows the position of the crank in relation to that of the eccentric. The face of the flange F next to the sheave is machined so as to leave a broad parallel projection n on it which fits into a corresponding recess on the back of the sheave; this forms the driving connection between the flange and the sheave. The sheave may be moved across the flange F in the direction C_1C_2, at right angles to the crank, within the limits of the two slots, and when in position is secured by the bolt and nut as shown. The distance ON is the same for all positions of the sheave and since ON is equal to the outside lap plus the lead it follows that the lead is constant. When the sheave is shifted its eccentricity is altered and so is its angular advance. By means of valve diagrams drawn for different positions of the eccentric it will be found that the nearer the centre of the sheave is to the point N then the earlier will be the points of cut-off, release, compression, and admission.

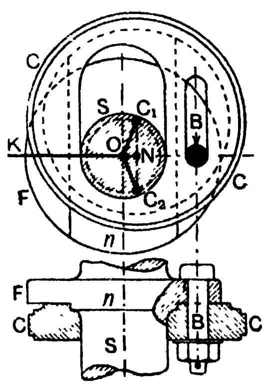


FIG. 464.

The position of the movable eccentric is sometimes controlled by a

powerful shaft governor, the bolt B being then absent. When so controlled the path of the centre of the sheave is generally an arc of a circle and the lead is then not constant.

260. Link Motions.—When a slide valve is driven directly by an eccentric it has been shown that the eccentric is in advance of the crank by an angle equal to $90° + \theta$ where θ is the angle of advance of the eccentric. The engine may be reversed by shifting the eccentric, but by using two eccentrics, one for forward and the other for backward motion, in conjunction with a simple and ingenious mechanism forming what is called a *link motion*, a much more convenient arrangement is obtained. The link motion is used not only for reversing an engine but also for varying the power developed by it, by altering the points of admission, cut-off, release, and compression. There are three principal types of link motion, the *Stephenson*, the *Gooch*, and the *Allan*, which will now be described.

261. Stephenson Link Motion.—The link motion most commonly used is the *Stephenson link motion* which is shown in Fig. 465. K is the

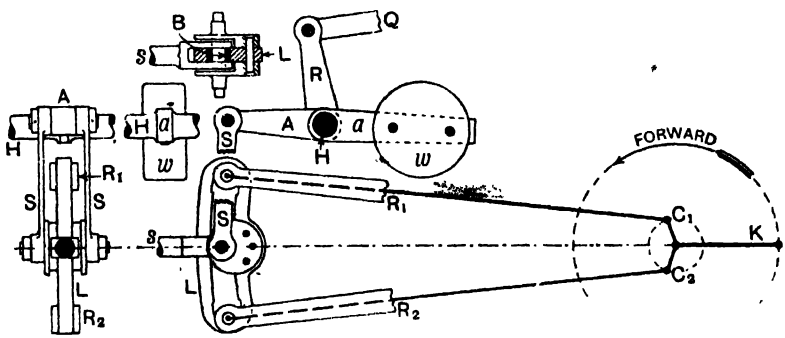


FIG. 465.—Stephenson link motion.

crank and C_1 and C_2 are the forward and backward eccentrics respectively which are fixed side by side to the crank shaft. The eccentrics are connected by eccentric rods R_1 and R_2 to the piece L called *the link*.

The link L is suspended by suspension rods S, one on each side of the link, from the end of the arm A which may be keyed, but is best forged, to the reversing shaft H. A weight w on the arm a, also attached to the reversing shaft, counterbalances the torque on H due to the weight of the link and eccentric rods. R is the reversing lever attached to the reversing shaft and operated from the cab through the rod Q. A rod s which is an extension of the slide valve spindle is forked to clear the link and carries a block B which fits into a slot in the link which is curved concave towards the crank shaft. The radius of the curved centre line of the slot in the link is equal to the length of the eccentric rods, the length of an eccentric rod being measured from the centre of the eccentric sheave to the point where the centre line of the rod cuts the centre line of the slot in the link.

When the link L is moved so that the outer end of the eccentric rod R_1 is as near as it can be to the block B the valve is practically operated by the forward eccentric, the backward eccentric having then

little influence on the motion of the valve, and when the link is moved so that the outer end of the eccentric rod R_2 is as near as it can be to the block B the valve is practically operated by the backward eccentric, the forward eccentric having then little influence on the motion of the valve. For intermediate positions of the link the valve has a reduced travel its motion being influenced by both eccentrics, the eccentric whose rod is nearest to the block B having the greater influence. It will be pointed out later that for intermediate positions of the link the motion of the valve is approximately that due to a single imaginary eccentric having an eccentricity and an angle of advance depending on the position of the link in relation to the block.

Various designs of link are used apart from the features which distinguish the Stephenson, Gooch, and Allan link motions from one another, and there are also differences in the positions of the joints of the link and eccentric rods and also in the position of the joint of the link and the suspension rods. In Fig. 465 the link is a *slotted link* and the joints of the link and eccentric rods are on the centre line of the slot in the link.

Fig. 466 shows a slotted link in which the joints between the link and the eccentric rods are placed behind the slot of the link. The advantage of this design is that the outer ends of the eccentric rods may be brought exactly opposite to the link block which is not the case with the design in Fig. 465. Eccentrics of greater eccentricity are required for the design of link shown in Fig. 465 than for that shown in Fig. 466.

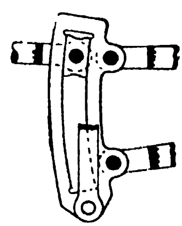


Fig. 466.

In Fig. 465 the suspension rods are jointed to the link at the centre of the latter while in Fig 466 the joint is at the bottom of the link. Longer suspension rods may be used with the design of Fig 466 which is an advantage, but with the design of Fig. 465 the motions of the valve in forward and backward gear are more nearly alike.

The form of link most commonly used in marine engine practice is

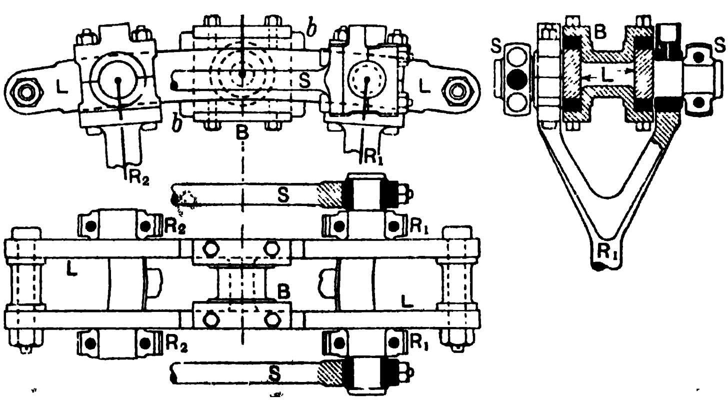


Fig. 467.—Marine engine link motion.

shown in Fig. 467. In this design the link consists of two curved steel bars L separated by distance pieces and bolted together at the ends as

shown. The block B is a steel forging, its centre forming the pin for the joint to the valve spindle. The block B is provided with gun-metal strips *b* which bear on the edges of the bars forming the link. The pins for the eccentric rods R_1 and R_2 and suspension rods S are forged on the link bars as shown.

262. Gooch Link Motion.—The difference between the Stephenson and the Gooch link motions is that in the former the link is shifted in relation to the block, which has no motion except in the line of the valve spindle, while in the latter it is the block which is shifted, the link being suspended from a fixed point.

The Gooch link motion is shown in Fig. 468. The parts which

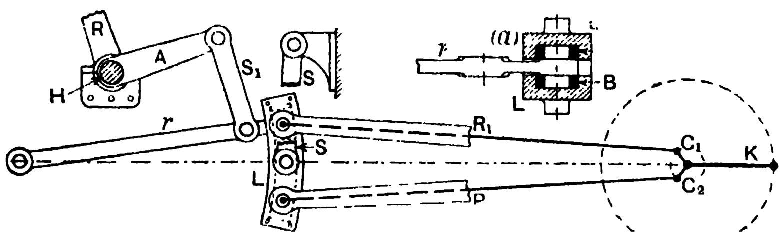


FIG. 468.—Gooch link motion.

have the same lettering as those in Fig. 465 have the same names and serve the same purposes.

The link L shown in Fig. 468 is of the box type but this type of link is not a necessary feature of the Gooch link motion. A box link consists of two curved bars of channel section bolted together at their ends. An enlarged cross section of the link and block is shown at (*a*). The block B is in two parts mounted on a pin fixed in one end of the radius rod *r*. The radius rod *r* is coupled to an extension of the valve spindle and is also suspended by the rod S_1 from the arm A on the reversing shaft H. By turning the reversing shaft the radius rod is made to swing and bring the link block into any position within the link. The radius of the curved centre line of the link is equal to the length of the radius rod. It will be observed that the Gooch link is convex towards the crank shaft whereas the Stephenson link is concave.

263. Allan Link Motion.—By arranging to move the link and the

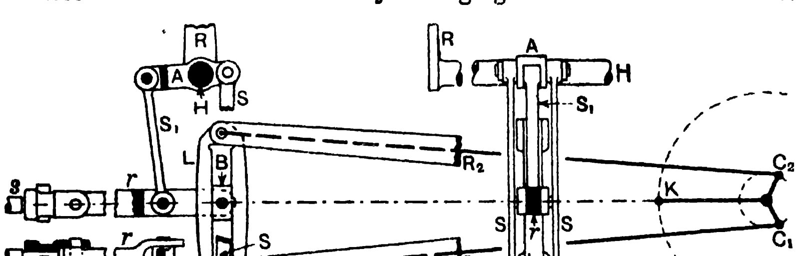


FIG. 469.—Allan link motion.

link block simultaneously in opposite directions when changing gear, the Allan link motion shown in Fig. 469 is obtained. The Allan link

motion may be looked upon as a combination of the Stephenson and Gooch link motions. The link, it will be seen, is straight and is suspended from one end of a lever A on the reversing shaft H while the radius rod r is suspended from the other end of that lever. By turning the reversing shaft the link and block are moved in opposite directions into their required relative positions.

264. Open and Crossed Eccentric Rods.—The crank being in its outer dead centre position, if the eccentric rods are as shown in Fig. 470 the arrangement is called one of *open eccentric rods*, while if

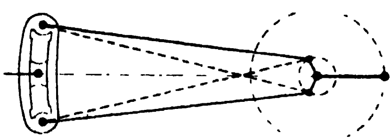


FIG. 470.

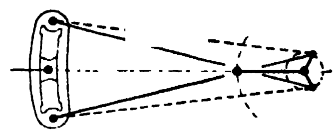


FIG. 471.

the eccentric rods are as shown by the thick dotted lines the arrangement is called one of *crossed eccentric rods*. These definitions do not imply that open rods do not become crossed or that crossed rods do not become open as the crank shaft rotates. In fact, as is shown in Fig. 471 where the crank shaft has made half a turn, the open rods become crossed and the crossed rods open, but in defining open and crossed rods the crank is placed as in Fig. 470.

265. Equivalent Eccentric.—If the motion of a slide valve whose line of travel is a prolongation of the line XOX, Fig. 472, is such that it is the sum of the harmonic motions due to two eccentrics OA and OB, which include a constant angle AOB, then the motion of the valve is the same as the harmonic motion due to an eccentric OC which is a diagonal of the parallelogram AOBC. This is proved as follows.

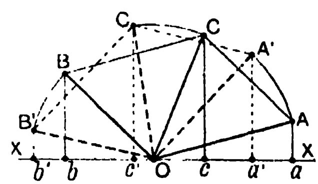


FIG. 472.

The lines Aa, Bb, and Cc are perpendicular to XX. Since equal parallel lines have equal projections on the same line, Oa, the projection of OA on XX, is equal to bc the projection of BC on XX. But $bc = b\mathrm{O} + \mathrm{O}c$. Therefore, $\mathrm{O}c = \mathrm{O}a - \mathrm{O}b$.

Let all three eccentrics be turned through equal angles in the same direction into the positions denoted by the accented letters. Then $\mathrm{O}a' = b'c' = b'\mathrm{O} - \mathrm{O}c'$. Therefore, $\mathrm{O}c' = \mathrm{O}b' - \mathrm{O}a'$. Hence,

$$\mathrm{O}c + \mathrm{O}c' = \mathrm{O}a - \mathrm{O}a' + \mathrm{O}b' - \mathrm{O}b, \text{ that is } cc' = aa' + bb'.$$

But cc' is the distance the valve is moved if driven by the eccentric OC, and $aa' + bb'$ is the distance the valve is moved if its motion is the sum of the motions in the line XX due to the two eccentrics OA and OB.

The eccentric OC is called the *equivalent eccentric* of the two eccentrics OA and OB.

If a long lever or link MN (Fig. 473) is made to swing about the

end N by means of an eccentric OA and a long eccentric rod AM, arranged as shown, and if a slide valve, whose line of travel is horizontal and passes through V, is connected to the lever at V, then the same motion will evidently be given to the valve by an eccentric *oa* connected to V by a long eccentric rod *a*V, arranged as shown, *oa* being parallel to OA and equal to OA $\times \frac{VN}{MN}$.

Again, if the valve is driven by making MN swing about the end M (Fig. 474) by means of an eccentric OB and a long eccentric rod

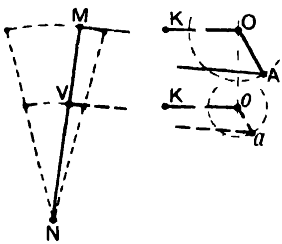


FIG. 473.

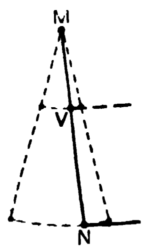


FIG. 474.

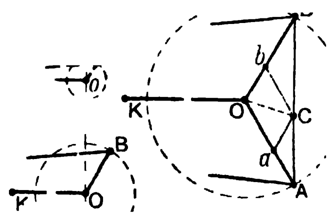


FIG. 475.

BN the same motion may be given to the valve by an eccentric *ob* connected to V by a long eccentric rod *b*V, *ob* being parallel to OB and equal to OB $\times \frac{VM}{MN}$.

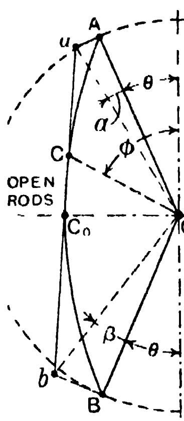
Now if the valve be driven by means of the link MN and the two eccentrics OA and OB acting simultaneously, the link MN being suspended so that its ends move under the action of the eccentrics in approximately horizontal lines, the motion of the valve will be the sum of the motions of the two eccentrics *oa* and *ob* applied directly to the valve. Applying the construction already given the eccentric OC (Fig. 475) equivalent to the two eccentrics O*a* and O*b* is found. If the eccentrics OA and OB be added in Fig. 475 and AB be joined it will be found that the point C lies on AB and divides AB so that AC : BC : : MV : NV.

Particular notice must be taken of the fact that the equivalent eccentric OC (Fig. 475) has been found on the assumption that the eccentrics OA and OB are on separate shafts rotating at the same speed in the same direction and at the levels of M and N respectively.

Coming now to the equivalent eccentric for the case of two eccentrics and an ordinary link motion, Figs. 476 to 479, the first step is to find the two eccentrics which, placed on separate imaginary shafts at the levels M and N, are equivalent to the eccentrics OA and OB respectively on the actual shaft. Keeping the crank OK horizontal and raising or lowering the eccentric OA until the line MO is horizontal will alter the angle of advance of OA from θ to $\theta \pm a$, the plus sign being taken for open rods and the minus sign for crossed rods. Also,[1] since the travel of M, due to the eccentric OA on the actual shaft, is greater than twice OA, the eccentric to take the place of OA at the level of M will require a greater eccentricity.

[1] See the Author's "Applied Mechanics," Art. 269.

In Figs. 476 and 479 Oa is the eccentric which, placed at the level of M, will be equivalent to the eccentric OA on the actual shaft. The length of Oa is obtained very approximately by drawing Aa at right angles to OA. The eccentric Ob which, placed at the level of N, will be equivalent to the eccentric OB on the actual shaft is obtained in



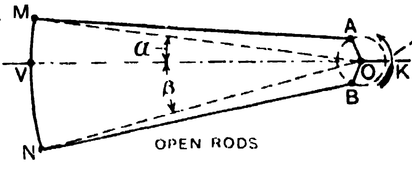
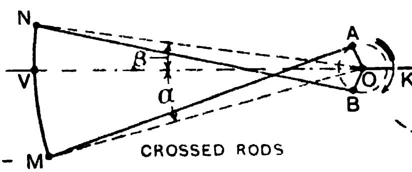


Fig. 477.



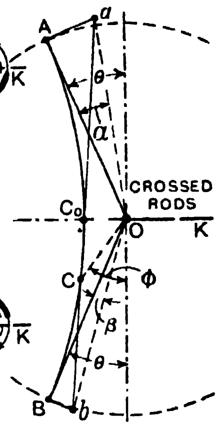


Fig. 476. Fig. 478. Fig. 479.

the same way, the angle of advance being $\theta \pm \beta$. Joining ab and dividing it at C so that aC : bC :: MV : NV, as in Fig. 475, the required equivalent eccentric OC is obtained, the length OC being its eccentricity and the angle ϕ its angle of advance. If the construction be repeated for different positions of the link MN it will be found that the locus of C is very approximately an arc of a circle ACB and C divides this arc so that AC : BC :: MV : NV. The arc ACB may be readily drawn after finding one intermediate position of C, preferably the position C_0 for mid gear.

In Mr. Macfarlane Gray's construction for finding the equivalent eccentric, the arc ACB is drawn with a radius equal to $\frac{AB \times AM}{2MN}$ and the arc is then divided at C so that AC : BC :: MV : NV.

It will be observed that for open rods the arc ACB is concave towards the crank shaft while for crossed rods it is convex.

Having found the equivalent eccentric, a Zeuner or other valve diagram may be drawn and the critical points in the operation of the valve determined.

Since the projection of OC (Figs. 476 and 479) on the horizontal line through O, the line of travel of the valve, is equal to the lead plus the outside lap, it is evident that in passing from full gear to mid gear the lead increases with open rods and decreases with crossed rods. It is also evident that when crossed rods are used the proportions of the gear must be altered to get approximately the same distribution of steam as with open rods, in particular the angle of advance θ must be considerably increased.

266. Radial Valve Gears.—In the various link motions which have been described, two points in the link receive motions from two separate eccentrics and the valve is connected to a third intermediate point whose position in the link is variable, such position deciding the direction of rotation of the crank shaft and the travel of the valve and consequently the ratio of expansion, compression, etc.

The common feature of what are called *radial valve gears* is a link, generally straight, in which one point M is made to move over a constant path, which is a closed curve, by means of an eccentric or crank either directly or through a combination of rods or links; another point N in the link is made to move over a path which may be varied, and to a third point V in the link the valve is connected. The points M, N, and V are at fixed intervals apart. In some gears V is between M and N while in others it is beyond N. It is by varying the path of N that the travel of the valve and the direction of rotation of the crank shaft are altered. Most commonly the point N is simply constrained to move in a straight line or in an arc of a circle and M is the only point in the link which is actually driven, and it is by varying the inclination of the guide of the point N that the travel of the valve and the direction of rotation of the crank shaft are changed.

267. Radial Valve Gears of the Hackworth Type.—The credit for the introduction of radial valve gears is generally given to J. W. Hackworth of Darlington who patented in 1859 the gear shown in Fig. 480.

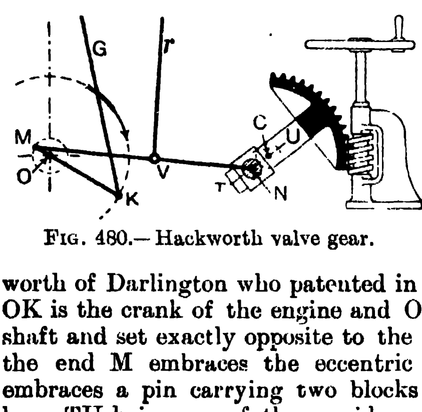


FIG. 480.—Hackworth valve gear.

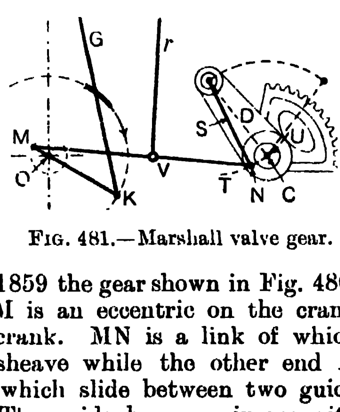


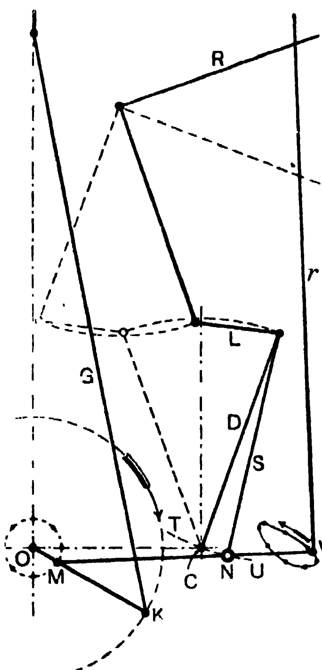
FIG. 481.—Marshall valve gear.

OK is the crank of the engine and OM is an eccentric on the crank shaft and set exactly opposite to the crank. MN is a link of which the end M embraces the eccentric sheave while the other end N embraces a pin carrying two blocks which slide between two guide bars, TU being one of these guides. The guide bars are in one with an axle, the whole being like a cranked axle, capable of turning in bearings of which C is the axis. The valve spindle is jointed to the upper end of the valve spindle link *r* whose lower end is jointed to the link MN at V. The inclination of TU may be varied by means of the worm wheel quadrant and worm shown. The travel of the valve is greatest when the inclination of TU to the line of piston stroke is least. As shown in Fig. 480 TU is set to full gear for clockwise motion of the crank shaft. At mid gear, for which the travel of the valve is least, TU is at right angles to the line of piston stroke. To reverse the direction of rotation of the crank shaft TU is inclined the other way.

The *Marshall valve gear* (Fig. 481) is a modification of the

Hackworth gear. The path of N is made an arc of a circle instead of a straight line. This is done by suspending N from the upper end of a radius arm D by a swinging link S. The inclination of TU, the path of N, is varied by changing the position of the radius arm which is capable of turning about the axis C. This turning may be effected by means of a worm wheel quadrant and worm or by other obvious means.

The *Bremme valve gear* differs from the Marshall gear in that the point V is beyond N instead of being between M and N; also the centre of the eccentric sheave is on the same side of the crank shaft as the crank. Fig. 482 is a centre line diagram of the gear, while Fig.



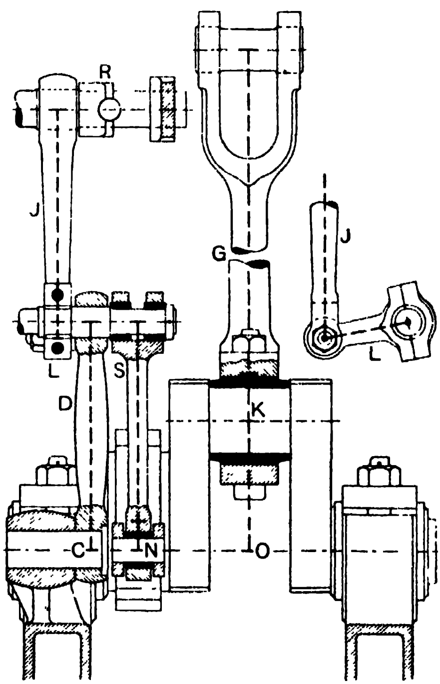


FIG. 482. Bremme valve gear. FIG. 483.

483 is a sectional elevation showing the forms of the various rods. For simplicity the crank is placed in the upper dead centre position in Fig. 483 and the radius arm D is in the vertical or mid-gear position. The gear is shifted by means of the reversing lever R, the arm J, and the drag link L.

Valve gears of the Hackworth type are generally proportioned so that, when the crank is in either dead centre position, N coincides with C. From this it follows that the position of the valve at the beginning of a piston stroke is not affected by the inclination of TU. Hence the lead for either stroke is constant, but of course the lead for one stroke need not be the same as for the other.

268. Joy Valve Gear.—Fig. 484 shows the *Joy radial valve gear* as applied to a locomotive. A point of resemblance of this gear to the Hackworth type is that the point N of the characteristic link MN is guided in an arc of a circle by means of a double guide TU similar to that used in the Hackworth gear (Fig. 480) except that it is curved instead of being straight. This guide is capable of turning in bearings about the axis C and, as in gears of the Hackworth type, it is by varying the slope of TU that the travel of the valve is altered and the direction of rotation of the crank shaft reversed. R is the reversing lever attached to the guide TU.

No eccentrics are used in the Joy valve gear. The motion of M is derived from a point E in the connecting rod G by being jointed to a link EF one end of which is jointed to the connecting rod at E while the other end is guided in an arc of a circle, which is approximately a

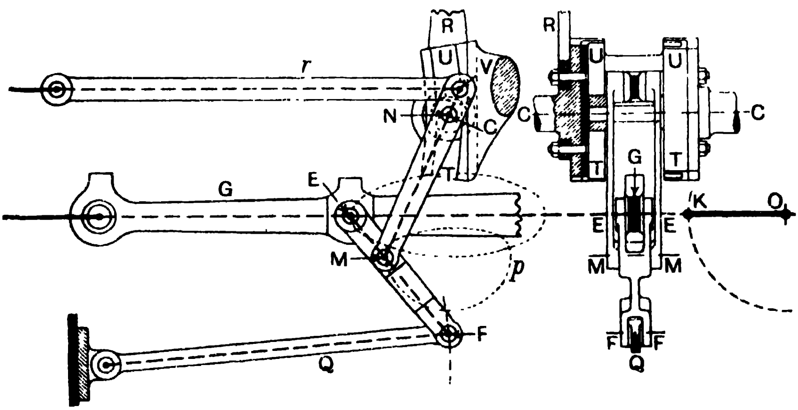


Fig. 484.—Joy valve gear.

vertical line, by means of the oscillating link Q. The path of M is roughly the same as would be obtained by means of an eccentric placed as in the Bremme gear. The actual path of M is the curve p whereas in the Bremme gear it is a circle.

In the Joy gear as in gears of the Hackworth type if the links are proportioned so that N coincides with C when the crank is in either dead centre position then the lead of the valve is constant for all inclinations of TU.

The radius of the curved centre line of TU is generally equal to the length of the valve spindle link r.

269. Walschaert Valve Gear.—Fig. 485 shows the *Walschaert valve gear* as applied to a locomotive. This gear has been extensively used on locomotives on the Continent of Europe and is becoming increasingly common in locomotive practice throughout the world.

The characteristic link MN is driven at M from the engine crosshead through the link HM. The point N in the characteristic link is driven by what may be described as a Gooch link motion with only one eccentric. In outside cylinder locomotives it is more convenient to use a return crank to form the equivalent of an eccentric. In Fig.

485 OC_1 the equivalent of an eccentric is formed in this way. The link L is a Gooch link supported in fixed bearings in brackets B as shown.

The variation in the travel of the valve is obtained by moving the radius rod r by means of the reversing lever R, the arm A, and the suspension link S, as in the Gooch link motion. Reversal of the direction of rotation of the crank shaft is effected when the link block carried by the rod r is moved to the other side of the axis about which the link oscillates.

The position of the eccentric OC_1 for constant lead is found as follows. The crank OK being in a dead centre position and the link L being so placed that the link block may be moved from one end of

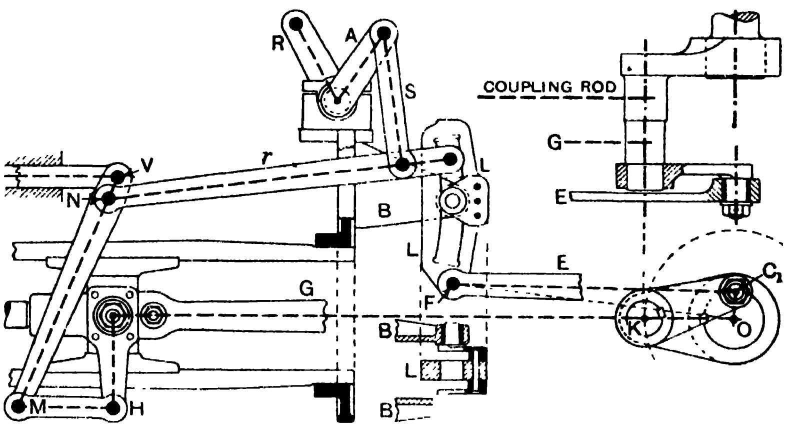


FIG. 485.—Walschaert valve gear.

the link to the other without moving the valve, then OC_1 is at right angles to OF.

270. Displacement Curves of Valve Gears—Correct Valve Diagram.—Drawing a centre line diagram of any valve gear and placing the crank in a considerable number of equidistant positions, preferably twenty-four, the loci of the various joints may be drawn and the actual displacement of the valve from its middle position for each numbered position of the crank may be determined.

Fig. 486 shows the curve traced by the lower end of the valve rod link of the Bremme gear shown in Fig. 482. The points numbered 1 to 12 correspond to the crank pin positions numbered 1 to 12 on the circle in Fig. 487. Referring to Fig. 486, YY is in line with the axis of the valve rod and from centres in this line and with a radius equal to the length of the valve rod link, arcs *ab*, *cd*, and XX are described. *ab* and *cd* are tangential to the curve and XX bisects *bd*. In the case illustrated the valve rod link is so long that the arcs *ab*, *cd*, and XX are nearly straight lines.

The distances of the points 1, 2, 3, etc. (Fig. 486), from XX measured in the direction parallel to YY are the distances of the valve from its middle position when the crank pin is in the positions 1, 2, 3, etc. (Fig. 487), respectively.

Taking the vertical diameter of the crank pin circle to represent the piston stroke, the corresponding positions of the piston are found in the usual way (Art. 250, p. 349) and from these the corresponding valve displacements are marked off at right angles to the line of piston stroke. A fair curve drawn through the points thus determined, together with straight lines parallel to the line of piston stroke

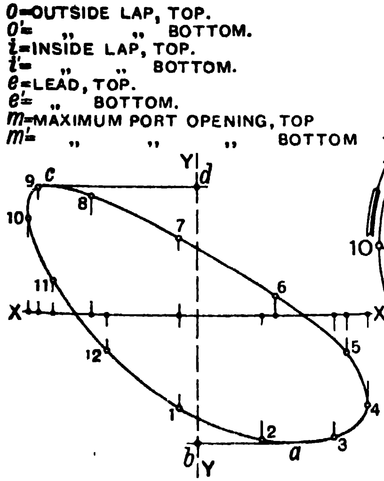


FIG. 486.

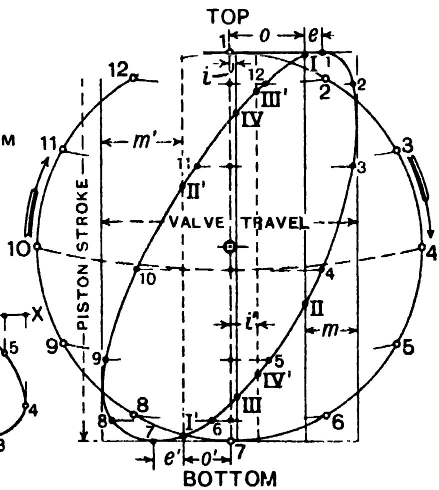


FIG. 487.

and at distances from it equal to o, i, $o + e$, etc., as shown, gives the *oval* or *elliptical valve diagram* from which the action of the valve may be fully determined. The points I, II, III, and IV give the piston positions at admission, cut-off, release, and compression respectively for the top of the piston, while I′, II′, III′, and IV′ are the same points for the bottom of the piston. Note that the inside lap for the top is negative in the case illustrated.

For a slide valve driven directly by a single eccentric the valve rod link becomes the eccentric rod and the curve of Fig. 486 becomes a circle whose radius is equal to the eccentricity of the eccentric. In that case if the obliquities of the eccentric rod and the connecting rod are neglected the oval shaped curve in Fig. 487 becomes a true ellipse called the *valve ellipse.*

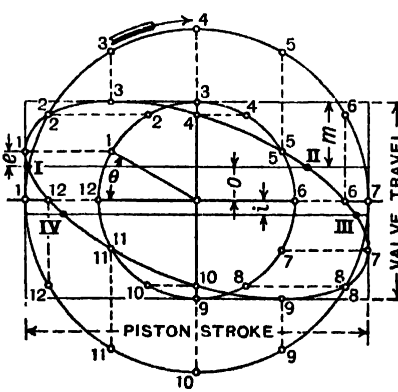


FIG. 488.—Valve ellipse.

A compact construction for drawing the valve ellipse is shown in Fig. 488. θ is the angle of advance of the eccentric, in this case 30°.

The points I, II, III, and IV give the piston positions at admission, cut-off, release, and compression, respectively, for one side of the piston. In practice the scale for the valve displacement should, in general, be full size while the scale for the crank pin circle should be considerably less than full size.

The oval or elliptical valve diagram is very suitable for testing a valve gear after it has been designed but in the preliminary design of the gear it is not so convenient as the other valve diagrams which have been described.

271. Equivalent Eccentric of Radial Valve Gears.—In Art. 265, p. 369, it has been shown how to determine the radius and angle of advance of an eccentric which will approximately give the same motion to a valve as two eccentrics driving the valve through a link MN. A similar construction is applicable to a radial valve gear.

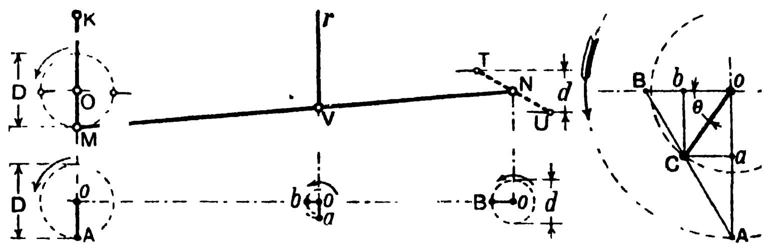


FIG. 489.—Equivalent eccentric of Hackworth gear.

First consider the Hackworth gear, Fig. 480, p. 372, and Fig. 489 above. The end N of the link MN is constrained to move in the straight line TU. Let T and U be the extreme positions of N for a given setting of the gear. The vertical travel of N is d and an eccentric oB having an eccentricity equal to $\frac{1}{2}d$ and placed under N and connected to it will give N the same vertical travel. The effect of the eccentric oB under N on the valve connected to the link MN at V will be approximately the same as that of an eccentric ob under V and connected to it, the radius ob being equal to $o\mathrm{B} \times \frac{\mathrm{MV}}{\mathrm{MN}}$ and ob being parallel to oB.

The vertical travel of the end M of the link MN is equal to D and the effect on the valve of the eccentric OM or oA under M will be approximately the same as that of an eccentric oa under V and connected to it, the radius oa being equal to $o\mathrm{A} \times \frac{\mathrm{NV}}{\mathrm{MN}}$ and oa being parallel to oA.

Since the motion of N in the line TU is due to the eccentric OM it is evident that, neglecting the obliquity of MN, when OM is vertical N will be at the middle of its travel and oB must be horizontal. Hence the eccentric oB which will give N the same vertical motion must be at right angles to OM or oA the eccentric which gives motion to M. To decide whether oB shall be to the right or left of o when OM is vertical it is only necessary to observe whether N is moving up

or down when M is moving from its vertical position. For anti-clockwise rotation and with K on the top dead centre as shown it is evident that N is moving downwards and *o*B must be to the left of *o*.

The single eccentric *o*C which is equivalent to the two eccentrics *oa* and *ob* is found as shown to the right in Fig. 489 to an enlarged scale. *o*C is a diagonal of the rectangle *oa*C*b* and θ is the angle of advance as explained in Art. 265. Instead of constructing the

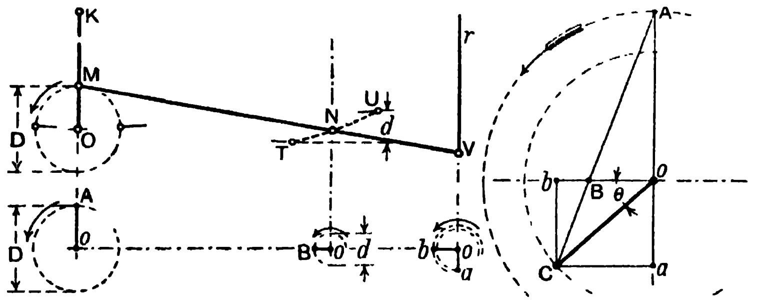


FIG. 490.—Equivalent eccentric of Bremmo gear.

rectangle *oa*C*b* to find *o*C, lengths *o*A and *o*B equal to the eccentricities of the eccentrics for M and N, respectively, may be set off at right angles to one another and, AB being drawn, C lies on AB, and AC : BC : : MV : NV.

The case of the Bremme gear is illustrated by Fig. 490 and is dealt with in the same way as the Hackworth gear but, since N now lies

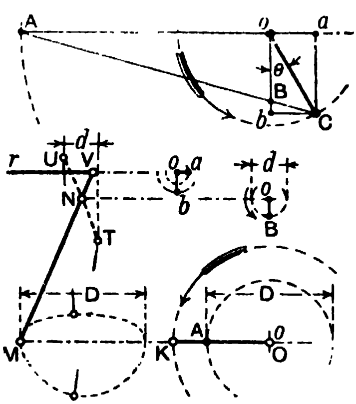


FIG. 491.—Equivalent eccentric of Joy gear.

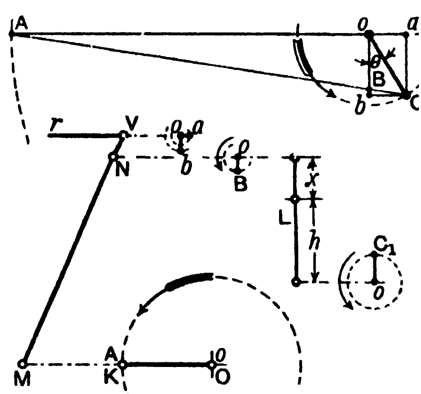


FIG. 492—Equivalent eccentric of Walschaert gear.

between M and V, *oa* and *o*A must point in opposite directions because when *o*A or OM is driving M downwards *oa* must be driving V upwards.

For the Joy gear the construction is the same as for the Bremme gear and is illustrated by Fig. 491. The curved path of M is shown but it is not necessary to draw this. The distance between the

positions of M corresponding to the dead centre positions of the crank K is the distance D, and oA is equal to $\frac{1}{2}D$. It will be sufficiently accurate to assume that T and U the extreme positions of N correspond to the crank positions which are at right angles to the line of piston travel.

In the Walschaert gear, Fig. 492, the travel of M in the direction of the line of piston travel is equal to the stroke of the piston and oA is equal to OK. The travel of N depends on the position of the link block in the link L. Let x be the distance of the link block from the fulcrum about which the link oscillates and let h be the distance of that fulcrum from F (Fig. 485, p. 375): then $oB = oC_1 \times \frac{x}{h}$. If x is above the fulcrum then oB and oC_1 point in opposite directions, but if x is below the fulcrum then oB and oC_1 point in the same direction. After finding oB the eccentrics oa and ob and the equivalent eccentric oC are found as before.

It will be seen that the approximate equivalent eccentric for a radial valve gear is easily determined and from it the action of the valve can be studied by means of a Zeuner or other valve diagram.

Working backwards the equivalent eccentric is useful in the preliminary design of a radial valve gear.

272. Drop Valves.—Double beat valves have been used to a considerable extent for distributing the steam in the cylinders of stationary engines. They have also been successful on locomotives on the Continent of Europe. The drop valve is particularly suitable for use with superheated steam and with it there is very little friction to oppose its motion, and since the valve is nearly balanced when under steam pressure the force required to operate it is comparatively small. The chief objection to the double beat valve is the difficulty of keeping the two seats simultaneously steam tight on account of differences in the expansion of the valve and seat with variations of temperature.

The application of drop valves to a stationary engine is illustrated by Fig. 494 which shows the high pressure cylinder and valve gear of a compound horizontal engine made by Messrs. George Saxon, Ltd., of Manchester, to whom the author is indebted for blue prints of the working drawings from which this illustration and also Figs. 493 and 495 have been prepared. Each cylinder has four valves, two for steam and two for exhaust, all of the double beat type with narrow, flat seats.

One of the exhaust valves is shown in detail in Fig. 493. V_2 is the valve and S its seat. Both valve and seat are made of cast iron, and to ensure, as far as possible, uniformity of material they are both cast from the same blowing of the cupola. Wings W are cast on the waist of the valve and these are inclined as shown so that the steam in passing the valve may turn it slightly, the valve being free to rotate on its spindle D. The spindle D passes through a long brass bush B, there being no stuffing-box. Leakage of steam past the spindle is prevented by the spindle being a good sliding fit in the long bush and by the addition of the circumferential grooves shown. The arrows show the direction of the flow of steam from the cylinder when the valve is open.

In the particular valve shown in Fig. 493 the outside diameter of the upper seat is 4·766 inches and the inside diameter of the lower seat is 4·25 inches, the width of the seats being 0·125 inch. The diameter of the valve spindle is 0·75 inch. If p is the absolute steam pressure, in lb. per square inch, in the inlet to the valve then the force pressing the valve on its seat is $0{\cdot}7854(4{\cdot}766^2 + 0{\cdot}75^2 - 4{\cdot}25^2)p = 4{\cdot}09p$ lb. Since the difference between the inside diameter of the upper seat and the outside diameter of the lower seat is only 1-64th of an inch, the upward force on the valve is negligible being only $0{\cdot}11p_1$ lb., where p_1 is the absolute pressure, in lb. per square inch, in the outlet from the valve. The force required to open the valve is

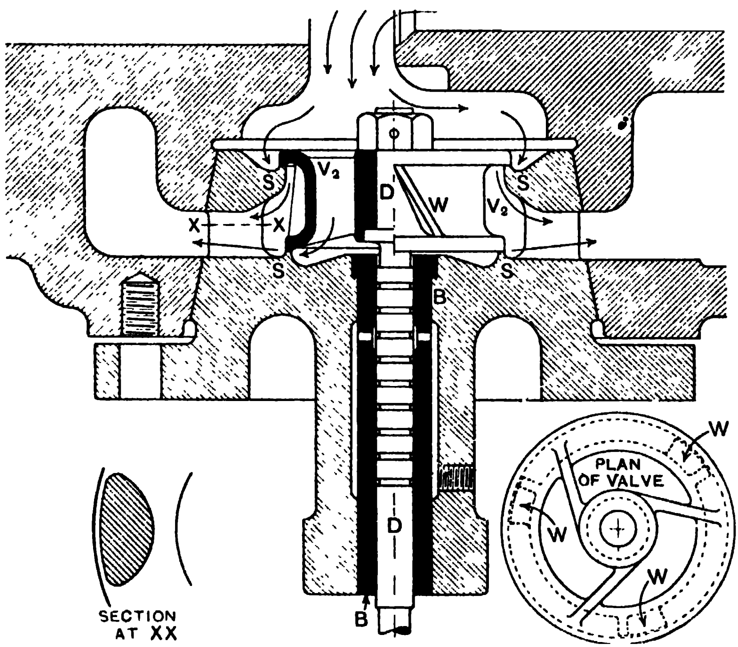


FIG. 493.—Double beat drop valve.

therefore $4{\cdot}09p$ plus the weight of the valve and the parts attached to it plus the force exerted by the spring to be presently referred to.

Referring now to Fig. 494, V_1 is the steam valve and V_2 the exhaust valve for one end of the high-pressure cylinder. The steam valve is opened by the eccentric C_1 through the eccentric rod R_1, the trigger T, and the lever L_1 turning on the fulcrum pin F_1, and is closed quickly by the helical spring H_1. The trigger T is mounted freely on the pin N which passes through the upper end of the eccentric rod R_1 and through the outer end of the link M which is mounted freely at its inner end on the pin F_1. When the valve is being lifted the hardened steel toe Q on the end of the arm A of the trigger T is in contact with the hardened steel pad P on one end of the lever L_1. The lifting of the valve continues until the arm B of

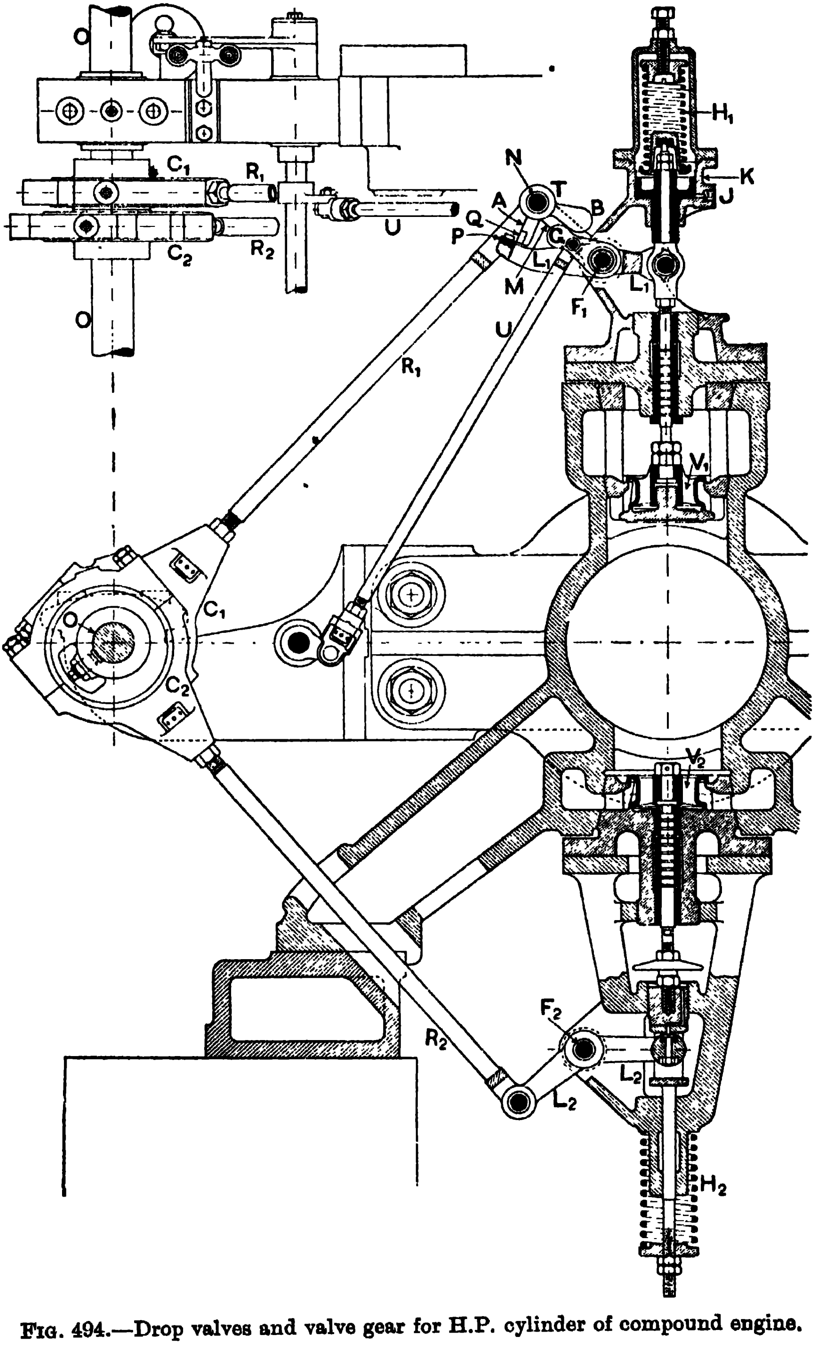


FIG. 494.—Drop valves and valve gear for H.P. cylinder of compound engine.

the trigger comes in contact with the arm G; the trigger then moves round and the toe Q moves inward and slips off the pad P. The lever L_1 is then free from the action of the eccentric, and the spring H_1 closes the valve.

The point at which the valve is liberated from the action of the eccentric is determined by the position of the arm G which is regulated by the governor through the rod U and other simple intermediate mechanism.

Enlarged views of the trigger T and the upper end of the eccentric rod R_1 are shown in Fig. 495. When not prevented by the arm G

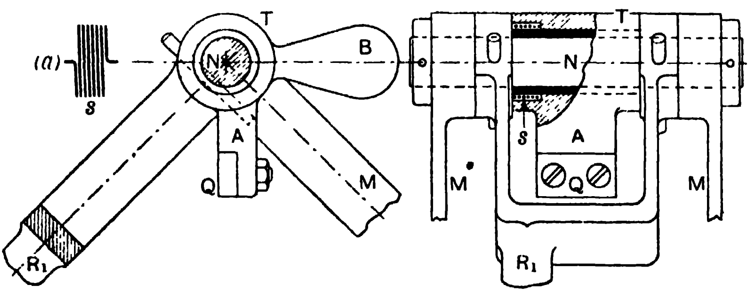


FIG. 495.

the trigger is moved into the position in which Q is over P by the weight of the arm B assisted by the torsion of a helical spring *s* which is housed in a recess in the boss of the trigger as shown. This spring is shown separately at (*a*). The ends of the wire forming the spring are bent outwards as shown and one projecting end enters a hole in the bottom of the recess in the trigger while the other is held in a short radial groove in the outer end of one prong of the fork of the eccentric rod.

To prevent the valve striking its seat violently a dashpot K is provided beneath the spring H_1. Into this dashpot fits a piston attached to the valve spindle. When the dashpot piston is rising as the valve is lifted air flows in through an adjustable orifice attached at J, and when the piston descends this air is compressed and discharged through the orifice which is adjusted to suitably retard the motion of the valve as it closes. In practice it is found that the air orifice can be adjusted so as to prevent all hammering action of the valve on its seat.

The exhaust valve V_2 is opened by means of the eccentric C_2 through the eccentric rod R_2 and the bent lever L_2 turning on the fulcrum pin F_2, and is closed by the helical spring H_2.

The eccentrics C_1 and C_2 are mounted on the shaft O which is driven from the crank shaft by equal bevel wheels.

The following particulars of the engine referred to above may be of interest. Diameter of H.P. cylinder, 11·5 inches. Diameter of L.P. cylinder, 25 inches. Stroke of pistons, 24 inches. Revolutions per minute, 120. The diameter of the steam valve for the H.P. cylinder is the same as that of the exhaust valve, namely, 4·5 inches, being the

outside diameter of the lower seat. The diameter of the steam valve for the L.P. cylinder is 8 inches, while that of the exhaust valve is 9 inches.

273. Corliss Valves.—An American engineer, Mr. George H. Corliss, introduced about 1850 a type of valve and valve gear which has been very largely used on mill engines in all parts of the world. The valve is an oscillating slide valve whose face is cylindrical, the axis of the valve being the axis about which it oscillates. Four valves are used on each cylinder, two for admission and two for exhaust. In the Corliss gear the valves are operated from a wrist-plate which receives an oscillating motion from an eccentric on the crank shaft. For the admission valves there is a trip gear, under the control of the governor, which gives a variable and sharp cut off.

Many gears have been devised to operate Corliss valves.

With Corliss valves the cylinder clearance volume is small.

Corliss valves are not favoured where superheated steam is used, and when operated through trip gears they are not suitable for high engine speeds, say, over 100 revolutions per minute.

Exercises XVIII

1. Longitudinal and transverse sections of the ports and slide valve seat and part of the valve are given in Fig. 496, the dimensions being in inches. The outside lap of the valve is 1 inch and the inside lap is 0·25 inch. The valve chest pressure is 180 and the exhaust pressure is 17, both in pounds per square inch absolute. What is the resultant force pressing the valve against its seat when it is open 0·25 inch to steam at one end? Neglect the weight of the valve.

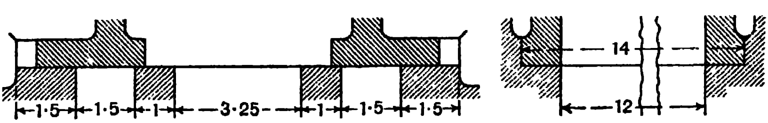


FIG. 496.

2. Keeping to the data of the preceding exercise but supposing that the valve is redesigned with a relief arrangement on the back such as is shown in Fig. 431, p. 346, the four Richardson strips enclosing a rectangular area of 6 × 9·75 square inches exposed to the exhaust pressure, and the strips presenting a rubbing surface of 17·75 square inches. What will then be the resultant force pressing the valve against its seat?

3. Taking the coefficient of friction between the valve and its seat as 0·09, calculate the pull or push on the valve spindle to overcome the friction of the valve as given in the two preceding exercises.

4. In a vertical marine engine there is a double-ported slide valve. The weight of the valve and part of the gear connected to it is to be balanced by means of a piston as in Fig. 430, p. 345. The weight to be balanced amounts to 1700 lb. The diameter of the valve spindle where it passes through the bottom of the valve chest is 3·25 inches. The valve chest pressure is 25 lb. per sq. in. absolute and the pressure above the balance piston is 5 lb. per sq. in. absolute. Compute the diameter of the piston.

5. In a slider crank mechanism the length of the connecting rod is four times the radius of the crank, which is 10 inches. Determine the displacement of the slider, in inches, for crank displacements of 30°, 60°, 90°, 120°, and 150° from the inner dead centre position. Also, what would the answers be if the obliquity of the connecting rod were neglected?

6. The travel of a slide valve is 5 inches and the angle of advance of the eccentric is 30°. The outside lap of the valve at the cover end is 1 inch and at the crank end 1·125 inches. What are the leads at the two ends?

7–12. In the exercises 7 to 12 given in the table below, r = radius of eccentric or half travel of valve, o = outside lap of valve, i = inside lap of valve, e = lead of valve, θ = angle of advance of eccentric, θ_1, θ_2, θ_3, and θ_4 are the angular positions of the crank at admission, cut-off, release, and compression, respectively, measured from the inner dead centre position. Linear dimensions are in inches and angles are in degrees. Determine the unknown quantities in each case so as to complete the table.

Ex.	r	o			θ	θ_1	θ_2		θ_4
7	2·75	1·25	0·375	0·25	?	?	?	?	?
8	2·75	1·375	0·125	0·125	?	?	?	?	?
9	2·625	1·625	0·5	?	45	?	?	?	?
10	2·625	?	?	0·125	?	?	90	140	?
11	?	2·75	0·25	?	?	−5	90	?	?
12	?	1·5	0·5	0·375	?	?	90	?	?

13. On the cover side of the piston the cut-off is at 0·8 of the stroke. The maximum opening of the port to steam is 1 inch and the lead is 0·3 inch. If the length of the connecting rod is twice the stroke of the piston find the travel of the valve and the outside lap.

14. On the crank side of the piston the lead of the valve is 0·5 inch, the maximum opening of the port to steam is 1·5 inches, the cut-off is at 0·7 and the release at 0·9 of the stroke. The length of the connecting rod is twice the stroke of the piston. Determine the travel of the valve and the outside and inside laps.

On the cover side of the piston the cut-off and release are the same as on the crank side. Find the outside and inside laps, the lead, and the maximum opening of the port to steam.

15. The following particulars relate to a Meyer expansion gear. Travel of main valve, 4 inches. Travel of expansion valve, 4 inches. Angle of advance of main eccentric, 30°. Angle of advance of expansion eccentric, 90°.

Find the angle which the crank makes with its initial dead centre position when the gap of the expansion valve is (1) + 1 inch, (2) 0, and (3) − 0·5 inch.

If the width of the ports in the main valve is 1 inch find the minimum width of each expansion plate so that their inside edges shall not overrun the inside edges of the ports in the main valve when the gap of the expansion valve is − 0·5 inch.

16. In a Meyer expansion gear the travel of the main valve is 4·25 inches, the travel of the expansion valve is 4·5 inches, the angle of advance of the main eccentric is 35°, and the angle of advance of the expansion eccentric is 90°. The length of the connecting rod is twice the stroke of the piston. Find the two gaps of the expansion valve for a cut-off at 40 per cent. of the stroke on both sides of the piston. If the plates are set with these gaps and the index plate is graduated from the gaps on the crank side of the piston, what will be the cut-off on the cover side of the piston when the pointer shows that the cut-off on the crank side is 20 per cent.?

17. A simple slide valve is driven directly by a movable eccentric such as is shown in Fig. 464, p. 365. The lead of the valve is 0·1 inch, outside lap 1 inch, inside lap 0·1 inch, and maximum travel 6 inches. Find the angular positions of the crank, measured from the initial dead centre position, at cut-off, release, compression, and admission, (*a*) when the valve has its maximum travel, and (*b*) when the travel is reduced to 3 inches by shifting the eccentric.

18. Referring to Figs. 476 and 477, p. 371, the dimensions of a Stephenson

link motion, with *open rods*, are as follows: $\theta = 13°$. OA = OB = 3·25 inches. AM = BN = 4 feet 3 inches. MN = 1 foot 5·5 inches. Find the eccentricity r and the angle of advance ϕ of the equivalent eccentric, and also the lead e of the valve, the outside lap being 1 inch, (1) at mid gear, (2) when MV = 6·25 inches, and (3) when MV = 3·75 inches (full gear).

19. Referring to Figs. 478 and 479, p. 871, the dimensions of a Stephenson link motion, with *crossed rods*, are as follows: $\theta = 25°$. OA = OB = 3·5 inches. AM = BN = 5 feet. MN = 1 foot 6 inches. Find the eccentricity r and the angle of advance ϕ of the equivalent eccentric, and also the lead e of the valve, the outside lap being 1 inch, (1) at mid gear, (2) when MV = 6·25 inches, and (3) when MV = 3·75 inches (full gear).

20. The particulars of a Bremme radial valve gear applied to a vertical engine and illustrated by Fig. 482, p. 373, are as follows: OC is horizontal and = $10\frac{3}{8}$ inches. OM = $1\frac{3}{4}$ inches. OK = 8 inches. Length of connecting rod G = 36 inches. MN = $10\frac{1}{2}$ inches. NV = $5\frac{3}{4}$ inches. Length of radius arm D = 14 inches. Length of swinging link S = 14 inches. Draw, full size, the locus of the point V for full gear and clockwise rotation of the crank shaft, the radius arm D being then inclined at 20° to the vertical. Next draw the oval valve diagram as in Fig. 487, p. 376, showing the valve displacement full size and the piston stroke one-quarter of full size. Allow for the length of the connecting rod but assume the length of the valve rod link r to be infinite. Take $e = 0·18$ inch, $e' = 0·31$ inch, $i = -0·1$ inch, and $i' = 0·2$ inch, and find the piston positions at cut-off and compression for both down and up strokes.

21. Draw, full size, the valve ellipse for a valve having a travel of $2\frac{1}{2}$ inches, the angle of advance of the eccentric being 25°. Take a piston stroke base 4 inches long. The laps of the valve are: outside, $\frac{7}{16}$ inch; inside $\frac{1}{8}$ inch. State the piston positions at cut-off and compression.

22. Taking the data given in exercise 20 find the eccentricity and angle of advance of the equivalent eccentric. Using this equivalent eccentric construct the oval valve diagram to the same scales as in exercise 20 allowing for the obliquity of the connecting rod but neglecting that of the valve rod link. Then by means of a tracing of one of the valve diagrams compare it with the other.

23. Take the following dimensions as applying to the drop valve shown in Fig. 493, p. 380. Inside diameter of upper seat, 4·52 inches. Inside diameter of lower seat, 4·25 inches. Diameter of central boss, 1·63 inches. Inside diameter of valve body, 3·25 inches. Thickness of central ribs, wings, and valve body, 0·19 inch. Calculate the lift of the valve so that the area of opening over the inner edges of the seats shall be the same as the area through the valve.

24. The seats of a double-beat valve have the following diameters, in inches. Upper seat, 7·9 and 7·5. Lower seat, 7·4 and 7. The absolute steam pressures in the inlet and outlet passages are 135 and 45 lb. per sq. in. Compute the force due to the steam pressures required to open the valve. Neglect the effect due to the spindle.

CHAPTER XIX

PERFORMANCE OF RECIPROCATING ENGINES

274. Engine Friction—Mechanical Efficiency.—An assumption which is frequently made is that the power consumed in overcoming the frictional resistances of a reciprocating steam engine is the same at all loads at constant speed. Making this assumption the horse-power required to overcome the frictional resistances, generally called the *friction horse-power*, for any load at a given speed is readily obtained by finding the indicated horse-power at no load at that speed. It would seem, however, from various tests, that the friction horse-power generally increases with the load, but not to any considerable extent.

It is now well established that if an engine is tested at various brake loads at approximately constant speed and if the indicated horse-power (I.H.P.) is plotted on a brake horse-power (B.H.P.) base, the points thus found practically lie on a straight line. It follows from this that since the friction horse-power (F.H.P.) at any given load is the difference between the I.H.P. and the B.H.P., the various points representing the F.H.P. on the same diagram will also lie on a straight line. Such a diagram is shown in Fig. 497.

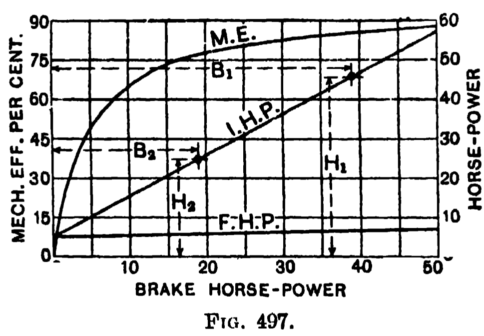


FIG. 497.

The equation to the I.H.P. line is $H = mB + F_0$, where H is the indicated horse-power corresponding to the brake horse-power B, F_0 is the friction horse-power at no load, and m is a coefficient which does not differ much from 1.

Since a straight line is fixed when two points in it are known, the I.H.P. line is determined when two trials are made at two different loads at the same, or nearly the same, speed. Let H_1 and H_2 be the I.H.P.'s corresponding to the B.H.P.'s B_1 and B_2.

Then, $H_1 = mB_1 + F_0$ and $H_2 = mB_2 + F_0$.

$$\text{Hence, } m = \frac{H_1 - H_2}{B_1 - B_2} \quad \text{and } F_0 = H_1 - mB_1 = H_2 - mB_2.$$

At a given load the ratio $\frac{\text{B.H.P.}}{\text{I.H.P.}}$ is called the *mechanical efficiency* of the engine at that load. If the mechanical efficiency be calculated and plotted for various loads the curve M.E., Fig. 497, is obtained. At full load the mechanical efficiency of reciprocating steam engines is generally between 80 and 90 per cent. In high-speed engines with forced lubrication the mechanical efficiency at full load may be as high as 96 per cent.

275. Mechanical Efficiency of Locomotives.—The friction horse-power of a locomotive, at constant speed, may be taken as the difference between the indicated horse-power and the draw-bar horse-power. This, however, includes the power required to overcome the frictional resistances of the tender, if there is one, as well as that of the locomotive itself.

The ratio of the draw-bar horse-power, at constant speed, to the indicated horse-power, may be called the mechanical efficiency of the locomotive.

The mechanical efficiency of express passenger locomotives is generally between 60 and 70 per cent. The tests on simple and compound goods locomotives, referred to in Art. 284, gave an average mechanical efficiency of 81 per cent. for the simple and 93 per cent. for the compound.

276. Available Heat supplied to an Engine.—The total heat H_1 per lb. of steam supplied to an engine is reckoned from 0° C. If the sensible heat of the feed water is h_1 per lb., reckoned from 0° C., and if this heat is recovered from the exhaust steam from the engine, then $H_1 - h_1$ is called the *available heat* in the steam, per lb., supplied to the engine.

The rule of the Inst. C.E. Committee on Engine and Boiler Trials is—"The available heat supplied to an engine is calculated as the total heat of the steam entering the engine less the 'water heat' of the same weight of water at the temperature of the engine exhaust, both quantities being reckoned from 32° F."

277. Steam Consumption and Equivalent Feed.—The performance of a steam engine is generally stated to be so many pounds of steam per horse-power (either I.H.P. or B.H.P. or E.H.P.) per hour. There is a vagueness about this way of stating the performance of an engine similar to that already considered in connection with the performance of steam boilers (Art. 156, p. 217). In connection with boiler performance the practice is now well established of giving the *equivalent evaporation from and at* 100° C. (212° F.) instead of, or in addition to, the actual evaporation under the actual conditions. A similar rule is wanted for steam engine performance.

The rule recommended by the Inst. C.E. Committee on Engine and Boiler Trials is to take the standard steam as that which has 1100 B.Th.U. (611 C.H.U.) available heat per lb. Hence, if W is the actual weight of steam, say per I.H.P. per hour, used under the actual conditions as to initial pressure and temperature and exhaust pressure, and if Q_B is the available heat in the actual steam in the actual engine in B.Th.U. and if Q_C be the same quantity in C.HU., then $\frac{WQ_B}{1100}$ or

$\frac{WQ_C}{611}$ is the equivalent steam consumption or *equivalent feed* as it is called.

The number 1100 (B.Th.U.) or 611 (C.H.U.) is very approximately the available heat in 1 lb. of dry saturated steam at an absolute pressure of 160 lb. per square inch when it is used in a compound condensing engine in which the exhaust pressure is 2 lb. per square inch absolute.

The relative economy of different engines, or the relative economy of the same engine working under different conditions, may be determined by reducing the steam consumptions to their "equivalent feeds."

A still more satisfactory way of stating the performance of an engine is to give the number of available heat units supplied, say per minute, per horse-power developed.

278. Thermal Efficiency.—The *thermal efficiency* of an engine is the ratio of the heat converted into work to the heat supplied. It is now usual to take the heat supplied to the engine as the "available heat" as defined in Art. 276.

EXAMPLE 1.—The steam is initially dry and saturated and its initial pressure is 180 lb. per square inch absolute. The final exhaust pressure is 3 lb. per square inch absolute and the engine uses 13·2 lb. of steam per I.H.P. per hour.

The total heat of dry and saturated steam at the initial pressure is 666·6 C.H.U. The sensible heat of water at the temperature corresponding to the exhaust pressure is 60·7 C.H.U. The available heat supplied is therefore 666·6 − 60·7 = 605·9 C.H.U per lb.

One I.H.P. per hour $= \frac{33,000 \times 60}{1400} = 1414{\cdot}3$ C.H.U.

Heat converted into work per lb. of steam $= \frac{1414{\cdot}3}{13{\cdot}2} = 107{\cdot}1$ C.H.U.

Thermal efficiency $= \frac{107{\cdot}1}{605{\cdot}9} = 0{\cdot}177.$

Or thus—

Heat equivalent of one I.H.P. per minute $= \frac{33,000}{1400} = 23{\cdot}57$ C.H.U.

Heat supplied per I.H.P. per minute $= \frac{605{\cdot}9 \times 13{\cdot}2}{60} = 133{\cdot}3$ C.H.U.

Thermal efficiency $= \frac{23{\cdot}57}{133{\cdot}3} = 0{\cdot}177.$

EXAMPLE 2.—The same as Example 1 except that the steam is initially superheated 100° C. and the steam consumption is 11·2 lb. per I.H.P. per hour.

Total heat in steam at initial pressure and superheated 100° C. = 721·8 C.H.U.

Available heat supplied = 721·8 − 60·7 = 661·1 C.H.U. per lb.

Heat supplied per I.H.P. per minute $= \frac{661{\cdot}1 \times 11{\cdot}2}{60} = 123{\cdot}4$ C.H.U.

Thermal efficiency $= \frac{23{\cdot}57}{123{\cdot}4} = 0{\cdot}191.$

279. Efficiency Ratio.—The ratio of the thermal efficiency as defined in the preceding Art. to the efficiency of the corresponding Rankine engine is called the *efficiency ratio.*

Referring to the examples worked out in the preceding Art. For the engine of Example 1 the efficiency of the Rankine cycle is 0·252 as in Case (*a*) of the example on p. 91.

$$\text{Efficiency ratio} = \frac{0{\cdot}177}{0{\cdot}252} = 0{\cdot}702.$$

For the engine of Example 2 the efficiency of the Rankine cycle is 0·260 as in Case (*c*) of the example on p. 92.

$$\text{Efficiency ratio} = \frac{0{\cdot}191}{0{\cdot}260} = 0{\cdot}735.$$

280. Economy due to Superheating.—The effect of superheating on the amount of heat required for a given power will be first studied by reference to two examples, the data and results of which are tabulated below. In the first example the initial pressure is assumed to be 100 lb. per square inch absolute while in the second it is 200 lb. per square inch absolute, and in both the back pressure is 3 lb. per square inch absolute. In each example there are two cases: (1) The steam is dry and saturated. (2) The steam is superheated 200° C. (360° F.).

Degrees of superheat	C.	0	200	0	200
	F.	0	360	0	360
Initial pressure	lb. per sq. in. abs.	100	100	200	200
Back ,,	,, ,, ,,	3	3	3	3
Rankine engine.					
Rankine cycle efficiency		0·218	0·240	0·257	0·279
Heat supplied per I.H.P. per min.	C.H.U.	108·1	98·2	91·7	84·5
	B.Th.U.	194·6	176·8	165·1	152·1
Saving due to superheating	p.c.	--	9·2	—	7·9
Saving due to increase of pressure without superheat	,,	-	—	15·2	—
Saving due to increase of pressure with superheat	,,	--	—	—	14 0
Actual engine.					
Heat supplied per I.H.P. per min	C.H.U.	157	130	135	115
	B.Th.U.	282·6	234	243	207
Saving due to superheating	p.c.	—	17·2	--	14·8
Saving due to increase of pressure without superheat	,,	—	—	14·0	—
Saving due to increase of pressure with superheat	,,	—	—	—	11·5
Specific volume at initial pressure	c. ft.	4·44	6·79	2·30	3·55
,, ,, back pressure	,,	118·5	118·5	118·5	118·5
Vol. per lb. after adiabatic expansion	,,	98·8	112·5	95 0	108·3
Heat drop per lb. ($H_B - H_C$, Art. 71, p. 87)	C.H.U.	130·9	168·2	156·2	199·3
	B.Th.U.	235·6	302·8	281·2	358·7
For equal heat drop.					
Weight	lb.	1·0	0 778	1·0	0·784
Volume at initial pressure	c. ft.	4·44	5·28	2·30	2·78
Volume after adiabatic expansion	,,	98·8	87·5	95·0	84·9

The Rankine cycle efficiency and the necessary heat supply per I.H.P. per minute have been worked out and tabulated. From these the saving due to superheating in the Rankine engine are then computed. The saving due to increase of pressure alone is also given. Then follow the probable corresponding figures for an actual engine.

The remaining part of the table will be referred to later.

In the Rankine engine it will be seen that the saving of heat due to superheating is 9·2 per cent. with the lower pressure and 7·9 per cent. with the higher pressure.

In the actual engine the probable saving due to superheating is 17·2 per cent. with the lower pressure and 14·8 per cent. with the higher pressure.

It now remains to be explained why superheating is so much more beneficial in the actual engine than in the perfect Rankine engine working between the same limits of pressure and temperature.

One great source of loss in the actual engine, and which has already been frequently referred to, is the transfer of heat from the steam to the cylinder walls during the early part of the stroke. Now if this transfer of heat results in the partial condensation of the steam, which will be the case if the steam is not sufficiently superheated to begin with, the moist steam will more readily part with heat to the cylinder walls than would dry or superheated steam. As will be shown presently a drop of hot water on the cylinder wall contains much more heat than the same volume of steam at the same temperature, and a slight fall in the temperature of that drop of water causes much more heat to be given to the cylinder wall than would be given by the same volume of steam for the same fall in temperature.

Superheated steam will suffer a certain fall in temperature and may be expanded a certain amount without condensing, the amount being greater the greater the superheat.

Another important reason for the greater economy of superheated steam over wet steam is that there is always a certain amount of leakage of steam past the valves and piston. Now leakage of wet steam means a greater loss of heat than a leakage of dry steam. Consider a pound of saturated steam at a pressure of, say, 100 lb. per square inch absolute. The total heat of this steam is 660·4 C.H.U. and its volume is 4·44 cubic feet. One pound of water at the same temperature contains 165·7 C.H.U. and its volume is 0·0178 cubic foot. The same volume of steam would contain $\frac{0\cdot0178}{4\cdot44} \times 660\cdot4 = 2\cdot65$ C.H.U. A given volume of this water therefore contains 62·5 times as much heat as the same volume of saturated steam at the same temperature. At lower pressures the ratio of the heat in the water to the heat in the same volume of steam at the same temperature is 105 to 1 at 50 lb. pressure and 326 to 1 at 10 lb. pressure.

Coming now to the lower part of the table on p. 389, it will be seen that the specific volume of the superheated steam is, for 200° C.

of superheat, over 50 per cent. greater than that of saturated steam at the same pressure. Based on this fact it is sometimes stated, that an engine using superheated steam must be larger than one using saturated steam for the same power, all other conditions being the same. A reference to the table shows, however, that this is not the case. In fact more work will be done in the same engine with a less weight of superheated steam than of saturated steam. What fixes the volume of an engine cylinder is the volume of the steam at the *end of the expansion* and not the volume at the *beginning.*

The student will find it a useful exercise to work out for himself the results given in the foregoing table.

281. Consumption of Superheated Steam at Reduced Loads.—An instructive diagram is given in Fig. 498 showing the effect of varying loads on the consumption of steam with varying amounts of superheat. This diagram was first published by Mr. Roger T. Smith in the *Proceedings of the Institution of Mechanical Engineers*, 1905, in the discussion on the "First Report to the Steam Engine Research Committee" by Professor David S. Capper.

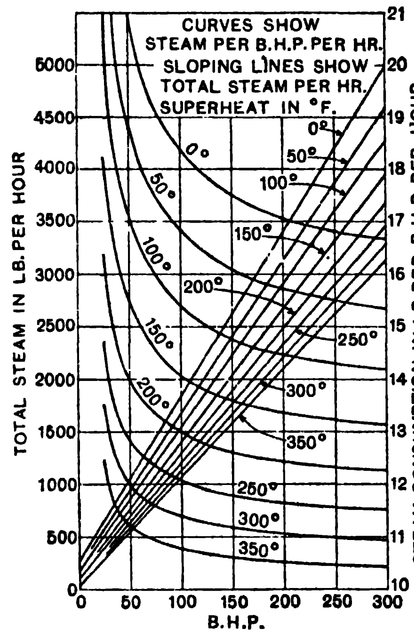


Fig. 498.

The engine to which Fig. 498 refers was a non-jacketed, quick-revolution, condensing, triple-expansion engine by Messrs. Belliss and Morcom. The full load of the engine was 300 B.H.P. It will be seen that the total consumption lines are "Willans lines." The economy due to superheating is clearly shown and the fact that the curves get flatter and more nearly horizontal as the superheat increases shows that the engine tended to become more nearly equally economical at all loads as the superheat increased.

282. Trials of a High-speed Triple-Expansion Engine.—The table on p. 392, given by Mr. A. E. Seaton in the *Proceedings of the Institution of Civil Engineers*, vol. cxcvi. (1914), p. 215, shows the results of careful trials made with a Belliss-Morcom triple-expansion engine. These results show not only the general economy of superheated steam but also the effect of superheating on the mean pressures in the cylinders, and on that in the condenser, the loads, the rates of revolution, and the speeds of the air and circulating pumps being similar in both trials.

Trials of a Belliss-Morcom Triple-Expansion Engine

(1) With saturated steam, (2) with superheated steam.

Cylinders, $13\frac{1}{2}$ inches, $20\frac{1}{2}$ inches, and 30 inches in diameter, by 14 inches stroke; speed, 350 revolutions per minute.

		(1)	(2)
Boiler pressure during trials	lb. per sq. in.	180	177
Steam pressure at H.P. cylinder valve-box	lb. per sq. in.	173	172
Steam temperature at H.P. cylinder valve-box		385° F.	500° F.
Vacuum in condenser	inches	24·8	26·2
E.H.P. steady load throughout		537	537
I.H.P. High pressure cylinder		222·0	233
I.H.P. Intermediate pressure cylinder		186·5	183
I.H.P. Low pressure cylinder		208·0	196
Total I.H.P.		616·5	612
Steam consumption per kilowatt-hour	lb.	23·4	18·26

Saving of steam due to superheating, 22 per cent.

From the temperatures and pressures in the H.P. cylinder valve-box given above it appears that there was a slight superheat, about 15° F., in trial (1) and a superheat of about 130° F. in trial (2).

The available heat per lb. of steam in (1) is about 1102 B.Th.U., while the available heat per lb. of steam in (2) is about 1177 B.Th.U.

The equivalent feed in (1) is $\frac{1102}{1100} \times 23{\cdot}4 = 23{\cdot}44$ lb. per kw. hour.

" " " (2) is $\frac{1177}{1100} \times 18{\cdot}26 = 19{\cdot}54$ " "

The real saving is $\dfrac{23{\cdot}44 - 19{\cdot}54}{23{\cdot}44} \times 100 = 16{\cdot}6$ per cent. instead of 22 per cent.

283. Superheated Steam in Locomotives.—The use of superheated steam in locomotives is now common. The degree of superheat is usually from 130° C. (234° F.) to 150° C. (270° F.) and the boiler pressure 180 lb. per square inch by gauge.

Superheating is more advantageous on long runs than on journeys with frequent stops. With express passenger engines the saving of coal per ton-mile due to superheating may be taken at about 20 per cent. With goods engines the saving is from 12 to 18 per cent.

284. Performance of Simple and Compound Locomotives.—Mr. George Hughes, Chief Mechanical Engineer, Lancashire and Yorkshire Railway, in his paper in the *Proceedings of the Institution of Mechanical Engineers*, 1910, on "Compounding and Superheating in Horwich Locomotives," has given much valuable information on the performance of locomotives. For the purpose of comparing the performance of simple and compound locomotives, tests were made on both under practically similar conditions.

Outside elevations of the simple and compound locomotives with their tenders are given in Figs. 499 and 500.

The following particulars relate to the boilers of both engines: Barrel, 4 ft. 10 in. diameter and 15 ft. long. Fire-box shell, 8 ft. 1 in. long and 4 ft. 1 in. wide. Copper fire-box, 7 ft. $2\frac{1}{4}$ in. long and 6 ft. $3\frac{3}{8}$ in. high. Tubes, 225 in number and 2 in. outside diameter.

Heating surface, 1767 sq. ft. in tubes and 147 sq. ft. in fire-box. Grate area, 23·05 sq. ft.

Figs. 499 and 500 show that both engines are of the 0–8–0 type. All the wheels are 4 ft. 6 in. diameter.

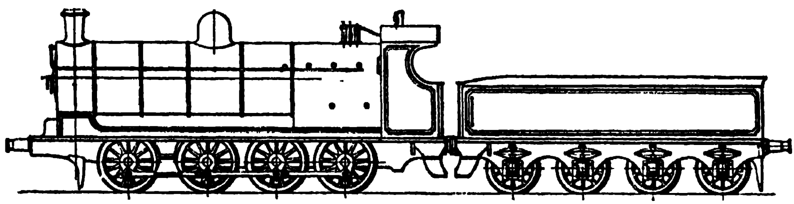


FIG. 499.—Two-cylinder simple locomotive.

The simple engine has two inside cylinders 20 in. diameter and a stroke of 26 in.

The compound engine has four cylinders—two outside H.P. cylinders $15\frac{1}{2}$ in. diameter and two inside L.P. cylinders 22 in.

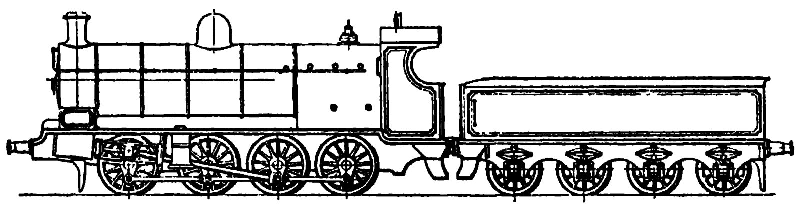


FIG. 500.—Four-cylinder compound locomotive.

diameter. All the pistons have a stroke of 26 inches. The sectional plan, Fig. 501, shows the arrangement of the cylinders and cranks. The valves are on top of the cylinders. The H.P. cylinders have piston valves and the L.P. cylinders have D slide valves.

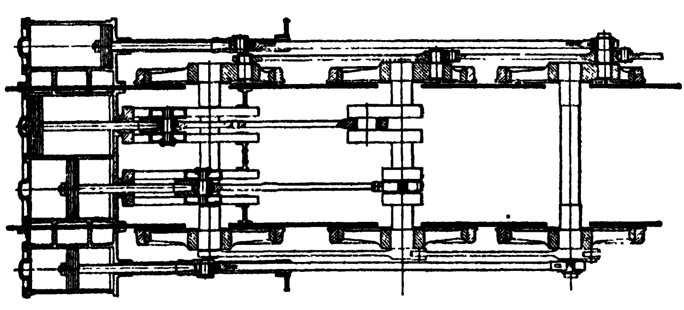


FIG. 501.—Four-cylinder compound locomotive.

An abstract of the average results of three trials of the simple and three trials of the compound locomotive is given in the following table which, while it shows the superior economy of the compound over the simple engine, contains figures on locomotive performance of interest to the student.

Averages of Tests of Simple and Compound Locomotives.

Particulars		Simple.	Compound.
Average steam pressure throughout test	lb. per sq. in.	178	180
Actual running time	min.	99·6	98
Average speed	miles per hour	21·9	22·3
Weight of train	tons	583·3	585·3
Weight of engine, tender, and train	tons	678·1	686·7
Number of wagons in train		59	58
Average indicated horse-power		701	546
Draw-bar pull	horse-power	567	506
Average draw-bar pull	tons	4·37	3·79
Average initial pressure in cylinders	lb. per sq. in.	140·1	148·9
Average steam pressure at release	lb. per sq. in.	55·1	28·4
Average percentage steam cut-off in cylinders		54	51·9
Coal burnt per sq. ft. of grate surface per hour	lb.	111·5	74·3
Average steam consumption per I.H.P. per hour	lb.	23·3	18·0
Average water consumption per I.H.P. per hour	lb.	26·9	22·8
Percentage saving of steam per I.H.P. from indicator diagrams			23·0
Coal per I.H.P. per hour	lb.	3·7	3·1
Percentage saving of coal per I.H.P. per hour			16

The coal used had a calorific value of 13,668 B.Th.U. (7593 C.H.U.).

285. Typical Performances of Various Types of Condensing Engines.—The following table gives the results of tests of various types of condensing engines, compound, triple-expansion, and quadruple-expansion, compiled from reliable sources.

Absolute initial pressure lb. per sq. in.	Superheat °C.	I.H.P.	Revs. per min.	Steam per I.H.P. per hour.	Heat supplied per I.H P. per min. C.H U.	Thermal efficiency.	Efficiency ratio.
			Stationary. Compound, condensing.				
164	50	1208	100	12·27	136	0·173	0·56
185	65	1400	80	11·74	129	0·183	0·645
161	95	1234	00	10·75	126	0·187	0·59
169	56	982	65	11·10	120	0·196	0·70
129	213	258	101	8·58	107	0·221	0·67
			Marine. Triple-expansion, condensing.				
180		645	61	13·35	139	0·171	0·59
160		1994	72	14·98	148	0·161	0·64
			Marine. Quadruple-expansion, condensing.				
214		9099	78	13·47	138	0·172	0·63

286. Steam Jackets.—Initial condensation in steam engine cylinders using saturated or wet steam may be prevented or reduced by the use of steam jackets. In estimating the performance of

an engine with jacketed cylinders it is most important that the amount of steam used in the jackets be included in the steam consumption.

A steam jacket is more beneficial the wetter the steam is on its admission into the cylinder, and if the steam is moderately superheated when admitted a jacket is of little or no advantage.

Initial condensation is greater the larger the clearance surface compared with the weight of steam in the cylinder, and it is generally more important to jacket the ends of the cylinder than to jacket the barrel. Jacketing is more advantageous for small cylinders than for large ones.

In single-cylinder engines, whether condensing or non-condensing, and in compound condensing engines, all using saturated steam, the saving due to jacketing may be 25 per cent. in small engines and less than 5 per cent. in large ones.

As would be expected the advantage of steam jackets is less the greater the rotational speed of the engine.

Steam jackets are more advantageous when an engine is running under reduced loads with earlier cut-offs.

Jacketing is more effective on the low-pressure cylinder of a compound engine than on the high-pressure cylinder.

The steam used in the jackets is steam at the initial working pressure.

It is important that all jackets be effectively drained so that there is no accumulation of water in them,

Whether jackets are used or not, the cylinders should be properly lagged to reduce as much as possible loss of heat by radiation. This also applies to steam pipes and receivers. There seems to be no advantage in having ordinary steam jackets on the receivers of compound and triple-expansion engines.

Exercises XIX

1. A full load test of a steam engine gave: I.H.P. = 167, and B.H.P. = 150. A half load test gave: I.H.P. = 91, and B.H.P. = 76, at the same speed. Compute the probable I.H.P., F.H.P., and mechanical efficiency when the load is 100 B.H.P.

2. The friction horse-power of an engine at full load is 30 per cent. greater than at no load at the same speed. The mechanical efficiency at full load is 90 per cent. Find the probable mechanical efficiency at half load at the same speed.

3. The mean pressure, in lb. per sq. inch, in the cylinder of an engine, when the boiler pressure and point of cut-off are kept constant, is expressed by the equation $p_m = 110 - 0{\cdot}07V$, where V is the piston speed, in feet per minute. Determine the piston speed, in feet per minute, which gives a maximum horse-power. Determine the maximum horse-power for a double-acting engine having a cylinder 20 inches in diameter. [U.L.]

4. Trials of a compound steam engine gave the following particulars :—

Revs. per minute	100	120	140	160	180	200	220
Brake horse-power	76	88	98	108	115	120	121·5
Indicated horse-power	86	100	112	123	133	142	147
Weight of steam used per hour . . . . . lb	1400	1530	1680	1830	1970	2120	2270

Plot curves showing the mechanical efficiencies and the steam consumptions at different speeds, and determine the speed at which the thermal efficiency on the brake is a maximum. Find the mechanical efficiency at this speed.
[U.L.]

5. In some recent trials of simple and compound locomotives the following particulars were obtained :—

Engine	Simple	Compound.
Weight of train behind locomotive	456 tons	487 tons.
Mean draw-bar pull	7750 lb.	8200 lb.
Weight of coal used	8500 lb.	6850 lb.
Weight of water used	61,000 lb.	49,500 lb.
Length of journey	82·5 miles	82·5 miles.
Time taken	98 min. 45 sec.	87 min. 2 sec.

Calculate the mean draw-bar H.P. in each case.

Assuming the calorific value of the fuel to be 12,000 B.Th.U. (6,667 C.H.U.), and the initial steam pressure 180 lb. per sq. inch by gauge, find the thermal efficiency of the locomotive, and the thermal efficiency of the boiler in each case.
[U.L.]

6. (a) During a trial of a triple-expansion engine the initial pressure was 200 lb. per sq. inch abs., and the initial superheat was 40° C. The exhaust pressure in the L.P. cylinder was 3 lb. per. sq. inch abs., and the steam consumption was 10·6 lb. per I.H.P. per hour.

(b) A compound engine on trial had a steam consumption of 11·9 lb. per I.H.P. per hour with an initial pressure of 165 lb. per sq. inch abs., and a back pressure in the L.P. cylinder of 3 lb. per sq. inch abs. The steam was initially dry and saturated.

Determine for these two trials: (1) The equivalent feed. (2) The available heat supplied per I.H.P. per minute. (3) The efficiency ratio.

7. A compound steam engine on trial was supplied with dry saturated steam at a pressure of 180 lb. per sq. inch abs. The weight of steam used per hour was 7200 lb. and the I.H.P. was 600. Taking the back pressure in the L.P. cylinder as 3 lb. per sq. inch abs., compute: (a) The available heat supplied per I.H.P. per minute. (b) The thermal efficiency. (c) The efficiency ratio.

8. In a trial of a triple-expansion engine the following observations were made :—

Steam pressure (absolute)	200 lb. per sq. inch.
Total heat per pound reckoned from water at 0° C. (32° F.)	669·8 C.H.U. / 1205·6 B.Th.U.
Weight of steam entering cylinder per hour	1200 lb.
Indicated horse-power	73·8

The steam entered a surface condenser and was condensed at a temperature of 56° C. (133° F.), and it was found that 24,000 lb. of condensing water per hour were raised through a temperature of 24° C. (43·2° F.). Determine the gross supply of heat per minute to the engine, the heat equivalent to the I.H.P., and the heat leaving the engine in the exhaust steam reckoned from 0° C. (32° F.).

Determine the amount not accounted for by radiation loss and errors of experiment, and show your results in the form of a balance sheet. [U.L.]

9. The following particulars relate to an engine trial made by Mr. Michael Longridge.

Type of engine.—Vertical cross-compound with overhung cranks.

Cylinders.—H.P., diameter, 20¾ inches. L.P., diameter, 45 inches. Diameter of piston rods, 5 inches. Stroke of pistons, 42 inches.

Data from observations.—Atmospheric pressure 14·7 lb. per sq. inch.
Weight of steam entering engine per hour, 8700 lb.
Pressure of steam by gauge at engine stop valve, 152·4 lb. per sq. inch.
Temperature of steam at engine stop valve, 497° F.
Temperature of air pump discharge, 115° F.
M.E.P. in H.P. cylinder from mean diagram, 68·2 lb. per sq. inch.
" " L.P. " " " " 9·18 " " " "
Revolutions per minute, 90·37.

From the foregoing particulars, and reference to steam tables, it is required to compute the following:—(1) I.H.P developed in H.P. cylinder. (2) I.H.P. developed in L.P. cylinder. (3) Total I.H.P. developed. (4) M.E.P. referred to L.P. cylinder. (5) Weight of steam used per I.H.P. per hour. (6) Heat supplied per I.H.P. per minute in B.Th.U. (7) Thermal efficiency. (8) Heat required by corresponding Rankine engine in B.Th.U. per I.H.P. per minute. (9) Efficiency ratio.

Note.—For (6) to (9) reckon heat from hot-well temperature 115° F.

CHAPTER XX

STEAM TURBINES

287. Action of Steam in Steam Turbines.—In a reciprocating steam engine work is done on a piston by the *statical* pressure of the steam and although the steam moves with the piston the kinetic energy of the steam plays an entirely neglible part in the working of the engine. In a steam turbine the steam moving with high velocity is made to impinge on numerous vanes, buckets, or blades on a wheel or rotor; the impinging steam exerts a *dynamical* pressure on the buckets and it is the kinetic energy of the steam which is converted into work.

When the steam reaches a steam turbine it has a large amount of heat energy, and it is at a pressure which permits of considerable expansion, but its kinetic energy is a negligible quantity, and before it can do work in the turbine some of its heat energy must be converted into kinetic energy; this is done by allowing the steam to expand in one or more properly designed passages without doing work, or, more correctly, work is done by the steam on itself by giving it velocity and therefore kinetic energy at the expense of some of the heat energy in the steam. This conversion of heat energy into kinetic energy may take place in one operation and the kinetic energy may then be applied, but in the majority of steam turbines the drop in pressure is made in a series of steps between which the kinetic energy generated in the previous step is applied. This means that there are a number of fixed elements used to convert heat energy into kinetic energy and an equal number of rotating elements placed alternately after the fixed elements, and in each rotating element the kinetic energy generated in the preceding fixed element is applied. A fixed element and its corresponding rotating element form a *turbine pair* or a *stage* of the turbine. In turbines of what is called the *reaction type* there is also a drop in pressure of the steam as it passes through the buckets of the rotor and the added kinetic energy is partly utilized in the buckets in which it is generated.

The first step in the study of the steam turbine is evidently the consideration of the conversion of heat energy of steam into kinetic energy.

288. Flow of Steam through an Orifice—Critical Pressure.—Suppose that a partition AB (Fig. 502) between a chamber C and a chamber D has an orifice O in it, the orifice being rounded as shown. Let the chamber C contain steam at a constant pressure P_1 lb. per

square foot or p_1 lb. per square inch, and volume v_1 cubic feet per lb. Also let the chamber D contain steam or air at a pressure P lb. per square foot.

Steam will flow from the chamber C through the orifice to the chamber D and the flow will be greater the smaller P is, provided that P is not less than a certain pressure, called the *critical pressure*, which lies between $0{\cdot}5P_1$ and $0{\cdot}6P_1$, and the discharge will be a maximum when P has the value of the critical pressure. For all values of P less than the critical pressure the discharge is the same as when P is equal to the critical pressure. These facts were first established experimentally by Mr. R. D. Napier in 1867.

FIG. 502.

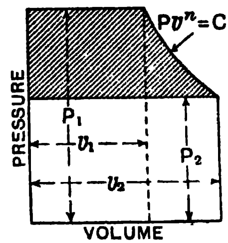


FIG. 503.

Let P_2 be the pressure of the steam, in lb. per square foot, in the throat, or narrowest section of the orifice, A the area of the throat in square feet, and V the velocity of the steam through the throat in feet per second. Also let v_2 be the volume of 1 lb. of the steam when it has expanded to the pressure P_2.

The flow or discharge W in lb. per second is $\dfrac{AV}{v_2}$.

In what follows it will be assumed that the pressure in the chamber D is equal to P_2, the pressure in the throat of the orifice.

If the flow is frictionless and adiabatic, the work done per lb. of steam as it expands in passing through the orifice is $\dfrac{P_1v_1 - P_2v_2}{n-1}$ ft.-lb., where n is the index in the equation $PV^n = C$. The steam in moving in the chamber C towards the orifice against the pressure P_1 does P_1v_1 ft -lb. of work per lb., and the work done against the pressure P_2 as the steam leaves the orifice is P_2v_2 ft.-lb. per lb. The net work done in giving velocity to the steam is represented by the hatched area in Fig. 503, and this will appear as kinetic energy in the steam as it leaves the orifice.

$$\text{Hence, } \frac{V^2}{2g} = \frac{n}{n-1}(P_1v_1 - P_2v_2) = \frac{n}{n-1}P_1(v_1 - rv_2) \text{ where } r = \frac{P_2}{P_1}$$

$$\text{Since } P_1v_1{}^n = P_2v_2{}^n, \quad v_2 = v_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}} = v_1\left(\frac{1}{r}\right)^{\frac{1}{n}}$$

$$\text{Therefore, } V = \sqrt{2g\frac{n}{n-1}P_1v_1\left(1 - r^{\frac{n-1}{n}}\right)}$$

$$\text{and, } W = \frac{AV}{v_2} = \frac{AVr^{\frac{1}{n}}}{v_1} = A\sqrt{2g\frac{n}{n-1}\frac{P_1}{v_1}\left(r^{\frac{2}{n}} - r^{\frac{n+1}{n}}\right)}$$

This makes W a maximum when $r^{\frac{2}{n}} - r^{\frac{n+1}{n}}$ is a maximum.

Let $y = r^{\frac{2}{n}} - r^{\frac{n+1}{n}}$ then $\frac{dy}{dr} = \frac{2}{n} r^{\frac{2-n}{n}} - \frac{n+1}{n} r^{\frac{1}{n}}$ and y is a maximum when $\frac{2}{n} r^{\frac{2-n}{n}} - \frac{n+1}{n} r^{\frac{1}{n}} = 0$, that is, when

$$r = \left(\frac{n+1}{2}\right)^{\frac{n}{1-n}} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

If the steam at the higher pressure is dry and saturated the value of n is about 1·135 and the value of r for maximum discharge is then 0·577.

If the steam at the higher pressure is sufficiently superheated to prevent condensation during expansion, the value of n is about 1·3 and the value of r for maximum discharge is then 0·546.

If the steam at the higher pressure is superheated but not sufficiently to prevent condensation throughout the whole of the expansion, the value of r for maximum discharge will lie between 0·577 and 0·546.

Inserting $r = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$ in the expression for W, the maximum discharge is

$$W_m = A\sqrt{2g\frac{n}{n-1}\frac{P_1}{v_1}\left\{\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right\}}$$

Putting $n = 1·135$, and $g = 32·2$,

$$W_m = 3·607A\sqrt{\frac{P_1}{v_1}} = 0·3007a\sqrt{\frac{p_1}{v_1}}$$

where a is the area of the throat in square inches, and p_1 is the initial pressure in lb. per square inch.

Putting $n = 1·3$, $W_m = 3·786A\sqrt{\frac{P_1}{v_1}} = 0·3155a\sqrt{\frac{p_1}{v_1}}$

Inserting $r = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$ in the expression for V then the velocity through the throat when the discharge is a maximum is

$$V_m = \sqrt{2g\frac{n}{n+1}P_1v_1}$$

Putting $n = 1·135$, $V_m = 5·851\sqrt{P_1v_1} = 70·21\sqrt{p_1v_1}$

Putting $n = 1·3$, $V_m = 6·033\sqrt{P_1v_1} = 72·40\sqrt{p_1v_1}$

The graphs of $V_m = 70·21\sqrt{p_1v_1}$ and $V_m = 72·40\sqrt{p_1v_1}$ are shown in Fig. 504, from which it will be seen that for initial pressures likely to occur in practice the range of the maximum velocity of steam in the throat of an orifice is comparatively small.

The mathematical demonstration which has been given proves that there is a certain critical pressure at the exit of the orifice for which the discharge of steam is a maximum. It follows that if the discharge

continues at that maximum value when the pressure just beyond the outlet is less than the critical pressure the conditions as to pressure and velocity within the orifice must remain constant, that is to say, while the pressure within the orifice at the outlet is the critical pressure, just beyond the outlet it may be very much less. This at first is difficult to realize, but the physical explanation is that to transmit a pressure through any medium requires time, and when the discharge is a maximum the velocity of the issuing steam is higher than the rate at which pressure can be transmitted through it. The lower pressure in the chamber D (Fig. 502) may be likened to a push against the jet of steam and if the jet is travelling faster than the rate at which the push can be transmitted through it the effect of the lower pressure can never reach the jet at the throat.

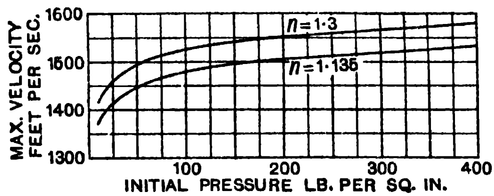


FIG. 504.

After the steam has left the orifice against a pressure less than that just within it the steam will of course expand to the lower pressure, but the kinetic energy produced by this further expansion will be reconverted into heat, the motion of the steam being now turbulent. The kinetic energy of a jet of steam issuing from an orifice is therefore very limited since the fall to the critical pressure is less than half the initial pressure. If, however, a properly designed divergent passage be added to the orifice on the outlet side a much larger amount of kinetic energy may be given to the jet and a larger pressure drop be utilized. This point is considered in the next Article.

The flow of steam through an orifice may also be studied with reference to the *heat drop* between the two extreme pressures within the orifice. The heat drop is the difference between the total heat of the steam per lb. at the pressure P_1 and the total heat at the pressure P_2. In estimating the total heat of the steam, allowance must of course be made for its condition as regards the amount of superheat, if any, or the amount of wetness if it is not dry.

Let the heat drop be denoted by U. Then for frictionless adiabatic flow JU is the increase in the kinetic energy of the steam per lb. due to the heat drop U. Hence neglecting the small initial velocity the steam may have, before entering the orifice, $\frac{V^2}{2g} = JU$, and

$V = \sqrt{2gJU}$.

If U is in C.H.U., then $V = 300\sqrt{U}$.

If U is in B.Th.U., then $V = 224\sqrt{U}$.

Also, using the same symbols as before, $W = \frac{AV}{v_2}$

The heat drop corresponding to the adiabatic expansion from one pressure to another is very readily found from the total heat-entropy chart.

EXAMPLE.—Let the pressures at the inlet and outlet of an orifice be 120 and 70 lb. per square inch respectively, and let the area of the throat be 1 square inch. It is required to find the velocity and discharge through the throat if the steam is dry and saturated at the higher pressure and the flow is frictionless and adiabatic.

First method.—$P_1 = 144 \times 120$, $P_2 = 144 \times 70$, $r = \frac{P_2}{P_1} = \frac{7}{12}$, $n = 1{\cdot}135$, $A = \frac{1}{144}$, and $v_1 = 3{\cdot}73$.

$$V = \sqrt{2g\frac{n}{n-1}P_1v_1\left(1 - r^{\frac{n-1}{n}}\right)}$$

$$= \sqrt{2 \times 32{\cdot}2 \times \frac{1{\cdot}135}{0{\cdot}135} \times 144 \times 120 \times 3{\cdot}73\left\{1 - \left(\frac{7}{12}\right)^{\frac{0{\cdot}135}{1{\cdot}135}}\right\}}$$

$= 1472$ feet per second.

$$W = \frac{AVr^{\frac{1}{n}}}{v_1} = \frac{1472 \times \left(\frac{7}{12}\right)^{\frac{1}{1{\cdot}135}}}{144 \times 3{\cdot}73} = 1{\cdot}705 \text{ lb. per second.}$$

Second Method.—The total heat of dry saturated steam at the higher pressure is 662·4 C.H.U. The total heat of dry saturated steam at the lower pressure is 656·6 C.H.U., its latent heat is 505·2 C.H.U., and its specific volume is 6·20 cubic feet.

Calculating the dryness fraction of the steam at the lower pressure after adiabatic expansion as explained in Art. 60, p. 69, it is found to be 0·964. Hence the heat of the wet steam at the lower pressure is $656{\cdot}6 - 505{\cdot}2 + 0{\cdot}964 \times 505{\cdot}2 = 638{\cdot}4$ C.H.U., and the heat drop is $662{\cdot}4 - 638{\cdot}4 = 24{\cdot}0$ C.H.U.

$V = \sqrt{2gJU} = \sqrt{2 \times 32{\cdot}2 \times 1400 \times 24} = 1471$ feet per second.

The actual volume of the wet steam at the lower pressure is $0{\cdot}964 \times 6{\cdot}20 = 5{\cdot}977$ cubic feet per lb.

$$W = \frac{AV}{v_2} = \frac{1471}{144 \times 5{\cdot}977} = 1{\cdot}709 \text{ lb. per second.}$$

The results obtained by these different methods are in sufficiently close agreement.

289. Flow of Steam through a Convergent-Divergent Nozzle.—It has been pointed out (p. 401) that when steam is issuing from an orifice against a pressure less than the critical pressure, the expansion which takes place after the steam leaves the orifice is not available for increasing the kinetic energy of the jet. It was also seen that the effective pressure drop in the orifice is less than half the initial pressure and that in consequence the kinetic energy of a jet issuing from an orifice is very limited. If, however, a divergent passage be made to form a continuation of the orifice the adiabatic expansion of the jet may be continued beyond the orifice and its velocity and kinetic energy may in consequence be considerably increased. The orifice and its divergent extension together form a convergent-divergent nozzle. Such

a nozzle is shown in Fig. 505 and its effect on the steam flowing through it will be best illustrated by a numerical example.

It will be assumed that the steam is initially dry and saturated and that the initial and exit pressures are 149 and 2 lb. per square inch respectively. It will also be assumed that the flow through the nozzle is frictionless and adiabatic and that the weight of steam passing through is 1 lb. per second. To satisfy these conditions the nozzle will require to have a certain definite diameter at the throat and another definite diameter at the exit. These diameters will first be determined.

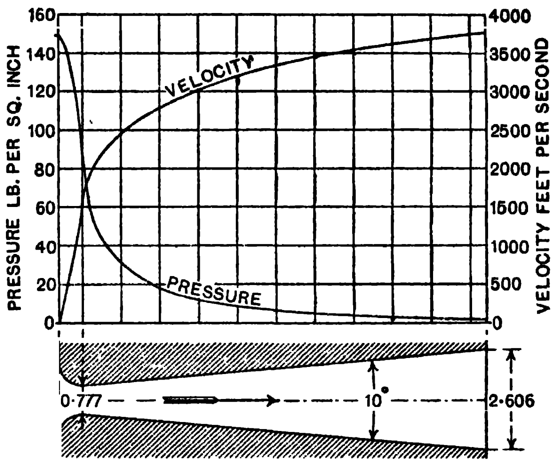


Fig. 505.

Considering the convergent part of the nozzle as an orifice, it follows that the pressure in the throat is the critical pressure which in this case will be $0·577 \times 149 = 86$ lb. per square inch. Computing the dryness fraction of the steam, after adiabatic expansion to a pressure of 86 lb. per square inch (Art. 60, p. 69), it is found to be 0·962. The specific volume of dry saturated steam at a pressure of 86 lb. per square inch is 5·11 cubic feet; therefore the actual volume of the wet steam at this pressure is $0·962 \times 5·11 = 4·916$ cubic feet. The total heat of this wet steam is $658·8 - 499·3 + 0·962 \times 499·3 = 639·8$ C.H.U.

The total heat in the dry saturated steam at the initial pressure is 664·6 C.H.U.

The heat drop is therefore $664·6 - 639·8 = 24·8$ C.H U.

Hence the velocity through the throat is $300\sqrt{24·8} = 1494$ feet per second.

Area of throat $= \dfrac{144 \times 4·916}{1494} = 0·4738$ square inch, and the diameter is 0·777 inch.

At the exit of the nozzle the pressure is 2 lb. per square inch. The dryness fraction after adiabatic expansion to this pressure is found to be 0·803. The specific volume of dry saturated steam at a pressure of 2 lb. per square inch is 173·5 cubic feet; therefore the volume of the wet steam at this pressure is $0·803 \times 173·5 = 139·3$ cubic feet. The total heat of this wet steam is

$$619·1 - 566·9 + 0·803 \times 566·9 = 507·4 \text{ C.H.U.}$$

The total heat drop is $664·6 - 507·4 = 157·2$ C.H.U.

Velocity at exit $= 300\sqrt{157 \cdot 2} = 3761$ feet per second.

Area of exit $= \dfrac{144 \times 139 \cdot 3}{3761} = 5 \cdot 333$ square inches, and the diameter is 2·606 inches.

Provided that the cross section of the nozzle changes gradually its form may vary considerably without any serious effect on its efficiency. A common form is shown in Fig. 505. The entrance up to the throat is rounded and beyond the throat the nozzle is conical, the angle of the cone being 10°. Adopting this form for the above numerical example and calculating the velocities and diameters for a number of intermediate pressures, the pressures and velocities may be plotted and the curves shown in Fig. 505 obtained.

290. Effect of Friction in a Nozzle.—All the steam passing through a nozzle does not follow perfect stream lines; there is a certain amount of eddying or irregular motion which results in friction between the molecules of the steam. There is also friction between the moving steam and the surface of the nozzle. All this friction has the effect of reducing the final kinetic energy of the jet, but the energy spent on friction is reconverted into heat which appears in the steam and increases its volume.

The effect of friction on the steam jet may be further explained by reference to steam charts, taking first the temperature-entropy chart, Fig. 506, and reproducing, for the sake of clearness, only the lines of the chart necessary for two particular cases. In one case the condition of the steam on entering the nozzle is represented by the point D, the temperature being t_1, pressure p_1, and dryness fraction BD/BC. Frictionless adiabatic flow to the lower temperature t_2 and pressure p_2 is represented by the vertical line DE, a line of constant entropy, and the dryness fraction at the end of this expansion, as the steam leaves the nozzle, is AE/AK. The heat returned to the steam, due to the friction, increases the entropy of the steam, and at the end of the expansion the increase of entropy is represented by EF, and the dryness fraction is then AF/AK, and the volume per lb. discharged is increased in the ratio of AF to AE.

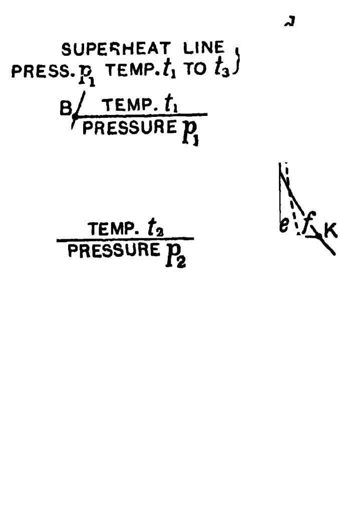


FIG. 506.

The heat in the steam per lb. discharged when there is friction is greater than when there is no friction by the amount represented by the hatched area EFLN, but when the whole of the available heat drop is given to the steam as kinetic energy this amount

of heat is represented by the area ABDE; therefore with friction the amount of heat converted into kinetic energy in the jet is represented by the area ABDE − the area EFLN.

The addition of entropy to the steam, when there is friction, goes on from the beginning to the end of the expansion so that the expansion line is a curve DF.

The case of steam initially superheated to the temperature t_3 and at the same pressure p_1 is also illustrated in Fig. 506. When there is no friction *de* is the expansion line and the heat converted into kinetic energy appearing in the jet as it leaves the nozzle is represented by the area ABC*de*, and when there is friction this amount is diminished by the amount represented by the area *efln* and the expansion line is the curve *df*.

Taking next the total heat-entropy chart, Fig. 507, and reproducing only the lines of constant pressure for the pressures p_1 and p_2 between which the expansion takes place in the nozzle. The condition of the steam on entering the nozzle is represented by the point D. Frictionless adiabatic expansion to the lower pressure p_2 is represented by the vertical line DE, a line of constant entropy. The length of DE, measured on the total heat scale, is equal to the total available heat drop. With friction the part DG is utilized to give kinetic energy to the jet. Hence, if the horizontal line GF be drawn to meet the constant pressure line p_2 at F, the point F represents the condition of the steam at the end of the expansion, and the expansion line is DF.

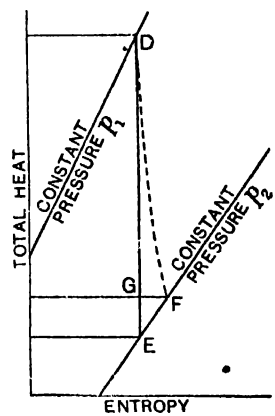


FIG. 507.

The ratio of the heat drop DG (Fig. 507) to the whole available heat drop DE is called the *efficiency of the nozzle*. The efficiency of the nozzle is also the ratio of the actual kinetic energy of the jet to the kinetic energy with frictionless adiabatic flow.

Mr. H. M. Martin gives [1] a formula for the heat loss in convergent-divergent nozzles, based on the results of experiments, namely, percentage heat loss $= 0{\cdot}06\,(u - 45)$, where u is the available heat drop in B.Th.U. If U is the available heat drop in C.H.U. the above formula becomes, percentage heat loss $= 0{\cdot}108\,(U - 25)$. For example, if U = 125 C.H.U. the heat loss is 10·8 per cent. or 13·5 C.H.U., and the efficiency of the nozzle is $\dfrac{125 - 13{\cdot}5}{125} = 0{\cdot}892$, or 89·2 per cent.

Experimental evidence goes to prove that in a convergent-divergent nozzle the loss due to friction in the convergent part up to the throat is practically negligible and all the loss may therefore be taken as occurring in the divergent part.

[1] "Design and Construction of Steam Turbines," p. 25.

Referring to the example worked out in Art. 289 and illustrated by Fig. 505, the effect of friction in that case may now be considered. The available heat drop was 157·2 C.H.U.

Using Mr. Martin's formula, the percentage heat loss due to friction is $0{\cdot}108(157{\cdot}2 - 25) = 14{\cdot}3$. The heat loss per lb. is therefore $0{\cdot}143 \times 157{\cdot}2 = 22{\cdot}5$ C.H.U.

The temperature of the steam at exit is 52·3° C. or $52{\cdot}3 + 273 = 325{\cdot}3°$ C. absolute. Referring to Fig. 506, but remembering that D coincides with C in the example now being considered, the increase of entropy due to friction is $EF = \frac{22{\cdot}5}{325{\cdot}3} = 0{\cdot}0692$.

$AE = 1{\cdot}5734 - 0{\cdot}1748 = 1{\cdot}3986$. $AF = 1{\cdot}3986 + 0{\cdot}0692 = 1{\cdot}4678$. $AK = 1{\cdot}9170 - 0{\cdot}1748 = 1{\cdot}7422$. The dryness fraction, with friction, at exit is $\frac{AF}{AK} = \frac{1{\cdot}4678}{1{\cdot}7422} = 0{\cdot}842$ instead of 0·803 without friction.

Volume of steam at exit $= 0{\cdot}842 \times 173{\cdot}5 = 146{\cdot}1$ c. ft.

The net heat drop is $157{\cdot}2 - 22{\cdot}5 = 134{\cdot}7$ C.H.U.

Velocity at exit is $300\sqrt{134{\cdot}7} = 3482$ feet per second instead of 3761 feet per second when there is no friction.

Area of exit $= \frac{144 \times 146{\cdot}1}{3482} = 6{\cdot}042$, and the diameter is 2·774 inches instead of 2·606 inches without friction.

291. Theory of the Injector.—The study of the steam nozzle leads to its application in the injector, an approximate theory of which may be considered here. A complete theory of the injector has not yet been formulated, and it must be understood that the formulæ obtained in what follows are to be regarded as approximate only.

The main features of the injector, as applied to feed-water into a boiler, are shown diagrammatically in Fig. 508. Steam from the same boiler, or from some other source, is supplied to the injector and passes through the convergent nozzle A and then into the convergent nozzle B. Between the nozzles A and B there is a gap in communication with a chamber C into which the feed-water enters. The steam and water mix in B, the steam being condensed. The water flows into the divergent nozzle E and then through a pipe to the boiler. A chamber D, provided with an outlet O, surrounds a gap between the nozzles B and E to allow of the escape of excess water at starting.

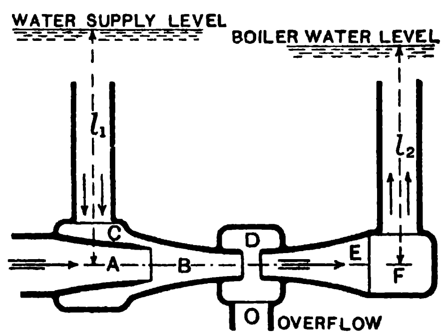


Fig. 08.

The object of the following investigation is to obtain formulæ for the weight of water taken into the injector per lb. of steam supplied, and the diameters of the throats of the nozzle for a given delivery

of water in a given time, also the probable temperature of the water delivered.

Let 1 lb. of steam at an absolute pressure of p_1 lb. per square inch and volume v_1 cubic feet per lb. be supplied per second.

The first step is to determine the velocity of the steam in the throat of the nozzle A. It is assumed that the steam expands adiabatically in the nozzle A. A reasonable assumption which is usually made is that the pressure in the throat of A is not less than the critical pressure (Art. 288), which for steam initially dry and saturated is $0{\cdot}577p_1$. It is further assumed that the pressure in the throat is $0{\cdot}6p_1$. It has been shown (p. 399) that the velocity in the throat of a convergent steam nozzle is—

$$V = \sqrt{2g\frac{n}{n-1}P_1v_1\left(1 - r^{\frac{n-1}{n}}\right)}$$

Putting $n = 1{\cdot}135$, $r = 0{\cdot}6$, and $P_1 = 144p_1$, $V = 67{\cdot}8\sqrt{p_1v_1}$.

Since v_1 diminishes as p_1 increases it will be found that for a considerable range of pressure V does not differ much from 1400 feet per second.

The velocity V may also be found by first determining the heat drop from the Mollier diagram or by calculation as already fully explained.

Let V_1 be the velocity of the water as it enters B from C, and let W be the weight of water entering C in lb. per second. Also let V_2 be the velocity of the water through the throat of B. $V_1 = \sqrt{2gl_1}$.

The momentum of the steam entering B $= \dfrac{V}{g}$. The momentum of the water entering B $= \dfrac{WV_1}{g}$. The momentum of the water in the throat of B $= \dfrac{(W + 1)V_2}{g}$

Hence, by the principle of the conservation of momentum,

$$\frac{V}{g} + \frac{WV_1}{g} = \frac{(W + 1)V_2}{g} \text{ or } V + WV_1 = (W + 1)V_2$$

If the water is lifted into C through a height l_1 by the action of the injector instead of being forced in as shown in Fig. 508, then $V - WV_1 = (W + 1)V_2$.

Hence, including both cases, $V_2 = \dfrac{V}{W+1} \pm \dfrac{WV_1}{W+1}$, and $W = \dfrac{V - V_2}{V_2 \pm V_1}$

Since V_1 is usually small compared with V the term $\dfrac{WV_1}{W + 1}$ may generally be neglected, then $V_2 = \dfrac{V}{W + 1}$ and $W = \dfrac{V}{V_2} - 1$

Owing to the presence of the gap between the nozzles B and E the pressure of the water in the throat of B may be taken as atmospheric, say 15 lb. per square inch, and as the water is warm its density may be taken as 62 lb. per cubic foot.

Kinetic energy of 1 lb. of water in the throat of B $= \dfrac{V_2^2}{2g}$

Pressure „ „ „ „ „ $= \dfrac{144 \times 15}{62}$

When the water reaches F its velocity has been considerably reduced by the divergent nozzle E and is now, say, V_3.

At F the pressure energy of 1 lb. of water is $\dfrac{144p_3}{62} + l_2$, where p_3 is the absolute pressure of the steam in the boiler in lb. per square inch, and l_2 is the head of water in feet at the delivery end of the injector.

Hence, by the principle of the conservation of energy,

$$\frac{V_2^2}{2g} + \frac{144 \times 15}{62} = \frac{144p_3}{62} + l_2 + \frac{V_3^2}{2g}$$

The water ultimately comes to rest in the boiler and the kinetic energy $\dfrac{V_3^2}{2g}$ may be taken equal to the pressure energy due to an addition of, say, 30 feet to the lift l_2 which corresponds to $V_3 = 44$ feet per second and to an addition of about 13 lb. per square inch to the boiler pressure.

Hence, $\dfrac{V_2^2}{2g} + \dfrac{144 \times 15}{62} = \dfrac{144p_3}{62} + l_2 + 30$

and, $V_2 = \sqrt{2g\left\{\dfrac{144(p_3 - 15)}{62} + l_2 + 30\right\}}$

Having found V and V_2 the weight of water W per lb. of steam may now be calculated from $W = \dfrac{V - V_2}{V_2 \pm V_1}$, or $W = \dfrac{V}{V_2} - 1$ if V_1 is neglected.

If the actual delivery is to be W′ lb. of water and w' lb. of condensed steam per second, then $W' + w' = W'\left(1 + \dfrac{1}{W}\right)$.

Let a_2 = area of throat of B in square inches, and
d_2 = diameter of throat of B in inches, then

$$a_2 = \frac{144W'\left(1 + \frac{1}{W}\right)}{62V_2}, \text{ and } d_2 = \sqrt{\frac{4a_2}{\pi}}$$

To reduce the velocity in the divergent nozzle E from V_2 to V_3 the diameter at the outlet should be $d_3 = d_2\sqrt{\dfrac{V_2}{V_3}}$.

Let v be the volume of the wet steam, in cubic feet per lb., after expansion in the nozzle A, let a_1 be the area of the throat in square inches, and let d_1 be its diameter in inches, then

$$\frac{a_1 V}{144} = w'v = \frac{W'}{W}v, \text{ and } a_1 = \frac{144W'v}{WV}, \text{ and } d_1 = \sqrt{\frac{4a_1}{\pi}}$$

Considering next the interchange of heat per lb. of steam.

Total heat in steam entering injector $= x_1 L_1 + h_1$, where x_1 is the dryness fraction, L_1 the latent heat, and h_1 the sensible heat.

Heat in water supplied $= Wh_2$.

Heat equivalent of kinetic energy in water supplied $= \pm \dfrac{WV_1^2}{2gJ}$.

Heat in (W + 1) lb. of water after mixing $= (W + 1)h_3$.

Heat equivalent of kinetic energy of water after mixing $= \dfrac{(W+1)V_2^2}{2gJ}$.

Hence, $x_1 L_1 + h_1 + Wh_2 \pm \dfrac{WV_1^2}{2gJ} = (W+1)h_3 + \dfrac{(W+1)V_2^2}{2gJ}$

an equation from which h_3 may be found.

The term $\dfrac{WV_1^2}{2gJ}$ is quite negligible and the term $\dfrac{(W+1)V_2^2}{2gJ}$ may generally be neglected. Neglecting these terms, the above equation becomes $x_1 L_1 + h_1 + Wh_2 = (W+1)h_3$.

Using Centigrade, h_2 may be taken equal to the temperature t_2 and h_3 equal to the temperature t_3. Using Fahrenheit, h_2 may be taken equal to $(t_2 - 32)$ and h_3 equal to $(t_3 - 32)$.

Exercises XX (*a*).

Note.—Pressures given are absolute unless stated to be otherwise.

1. The diameter of the throat of an orifice is 0·75 inch. Steam initially dry and saturated, and at a pressure of 150 lb. per square inch, flows through this orifice against a back pressure of 90 lb. per square inch. Compute the weight of steam discharged per second, assuming frictionless adiabatic flow. Determine also the critical back pressure and the maximum possible discharge.

2. Determine the diameter of the throat of an orifice through which there is a flow of 21 lb. of steam per minute. The initial pressure is 120 lb. per square inch and the back pressure is 72 lb. per square inch. The steam at the higher pressure is dry and saturated. Assume frictionless adiabatic flow.

3. Calculate the velocity of steam through the throat of an orifice, in feet per second, for maximum discharge, when the initial pressures are :—50, 100, 200, and 300 lb. per square inch. The steam is initially dry and saturated and the flow is frictionless and adiabatic.

4. Steam, dry and saturated, at a pressure of 170 lb. per square inch is supplied to a convergent-divergent nozzle and is delivered at a pressure of 2 lb. per square inch. Determine the diameters at the throat and exit of the nozzle if the delivery of steam is 30 lb. per minute. Assume frictionless adiabatic flow.

5. Same as exercise 4 except that instead of the steam being initially dry and saturated it is wet, the dryness fraction being 0·980. Also, express the kinetic energy of the jet at exit, in this exercise, in terms of that in exercise 4.

6. Steam flows through a properly designed nozzle and the pressure drops from 180 to 2 lb. per square inch. Assuming frictionless adiabatic flow, compute the dryness fraction and the velocity of the steam as it leaves the nozzle when the steam at the higher pressure is, (*a*) dry and saturated, (*b*) superheated 60° C. above the saturation temperature.

7. A convergent-divergent nozzle is required to pass 720 lb. of steam per hour with a pressure drop from 190 to 2 lb. per square inch. The steam at the higher pressure is dry and saturated. Assuming that the frictional resistance is equivalent to 13 per cent. of the available heat drop, determine the diameters at the throat and exit.

8. An injector is required to deliver 20 gallons of water per minute from a

tank whose constant water level is 4 feet below the level of the injector into a boiler in which the steam pressure is 200 lb. per square inch absolute. The water level in the boiler is 5 feet above the level of the injector. The steam for the injector is to be taken from the same boiler and it is to be assumed as dry and saturated. The temperature of the water in the supply tank is 15° C. Compute: (a) The weight of water taken from the supply tank per lb. of steam. (b) The diameter of the throat of the mixing nozzle. (c) The diameter of the throat of the steam nozzle. (d) The temperature of the water leaving the injector. Neglect radiation losses.

9. The same as exercise 8 except that the boiler pressure is to be 60 instead of 200 lb. per square inch absolute.

10. The same as exercise 8 except that the steam supplied to the injector is to be the exhaust steam from a non-condensing engine, this steam having an absolute pressure of 20 lb. per square inch and a dryness faction of 0·85.

292. Force and Momentum.—When the force of gravity acts on a mass of W lb. and the mass is quite free to move directly towards the earth, its velocity increases uniformly at the rate of g feet per second every second during which the force acts and the body moves freely. If the same mass is acted on by a force equal to that of gravity but in, say, a horizontal direction, and the mass is free to move in that direction, then its velocity will increase at the same rate as before, namely, g feet per second per second. Now suppose that the same mass of W lb. is acted on by a force of P lb. and that the mass is free to move in the direction in which P acts, then the velocity of the mass will increase at the rate of f feet per second per second, and $\frac{P}{W} = \frac{f}{g}$ or $P = \frac{Wf}{g}$. This is a fundamental principle of mechanics.

If a mass of W lb. has a velocity of V feet per second at any instant, then $\frac{WV}{g}$ is called the *momentum* of the mass at that instant.

Again, if a force of P lb. acts on a moving mass of W lb. for one second, and the velocity of the mass in the direction in which P acts changes from V_1 to V_2 feet per second, then $V_1 - V_2 = f$, and $P = \frac{Wf}{g} = \frac{W(V_1 - V_2)}{g} = \frac{WV_1}{g} - \frac{WV_2}{g}$, that is, change in momentum per second is equal to the force producing that change.

293. Action of Steam on Buckets of Impulse Turbine.—In a pure impulse turbine the buckets of a stage are surrounded by steam at the same pressure everywhere and there is therefore no drop in pressure as the steam flows through the bucket passages. The flow through the bucket passages is due to the kinetic energy of the steam, but in flowing through the absolute velocity of the steam is reduced and it gives up kinetic energy and in doing so causes a dynamic pressure or driving force on the buckets.

Referring to Fig. 509 AB is the axis of a nozzle which delivers steam at a high velocity on to a series of buckets mounted on the rim of a wheel or rotor. On the axis of the nozzle a length AB is taken to represent the velocity V_1 of the steam as it leaves the nozzle and impinges on the buckets of the wheel. CB represents in magnitude

and direction the mean velocity S of the buckets, and the line AC will represent the velocity R_1 of the steam in relation to the buckets. If the steam is to enter the bucket at B without shock AC must be the direction of the tangent to the bucket at entrance.

Completing the parallelogram ACBD, R_1 is also represented by DB. Fixing attention on the bucket at B, the steam will flow over its concave surface and leave it at b with a relative velocity R_2 in the direction of the tangent bd to the inner curve of the bucket at b. If

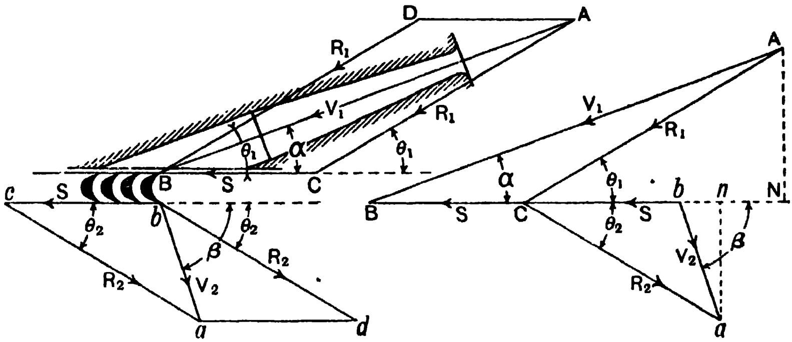


FIG. 509. FIG. 510.

there is no friction $R_2 = R_1$, but friction causes R_2 to be less than R_1. Making bd equal to R_2, and bc, parallel to CB, equal to S, and completing the parallelogram $dbca$ the absolute velocity V_2 of the steam as it leaves the bucket is represented by ba.

The essential lines in the velocity diagrams given in Fig. 509 may be combined in one diagram as shown in Fig. 510.

Another form of the velocity diagram is shown in Fig. 511; this is the most convenient form to use when dealing with compound turbines. In Fig. 511 the triangle of velocities at entrance 1VR corresponds to the triangle ABC in Fig. 510 and the triangle of velocities at exit 2VR corresponds to the triangle abc. The arrangment and lettering of the diagram in Fig. 511 will be more fully appreciated later when dealing with compound turbines.

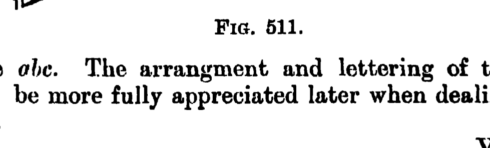


FIG. 511.

The kinetic energy of the steam per lb. as it enters the buckets is $\frac{V_1^2}{2g}$ and its kinetic energy as it leaves the buckets is $\frac{V_2^2}{2g}$. Hence, if there is no friction, the energy given to the wheel per lb. of steam is $\frac{V_1^2 - V_2^2}{2g}$. This may be called the *indicated work* on the wheel since

it corresponds to the indicated work in the cylinder of a reciprocating engine.

If P is the mean tangential effort on the wheel per lb. of steam per second, then $PS = \frac{V_1^2 - V_2^2}{2g}$ and $P = \frac{V_1^2 - V_2^2}{2gS}$.

The energy given to the wheel and the effort exerted on it may also be arrived at by considering the change in the momentum of the steam. On entering the buckets the velocity of the steam *in the direction of the motion of the buckets* is represented by NB (Fig. 510) which is equal to $V_1 \cos a$. This is called the *velocity of whirl* of the steam at entrance. On leaving the buckets the velocity of the steam in the direction of the motion of the buckets is represented by *nb* (Fig. 510) which is equal to $- V_2 \cos \beta$, and this is the velocity of whirl at exit.

In passing through the buckets the change in the momentum of the steam per lb. per second in the direction of the motion of the buckets is $\frac{1}{g}(V_1 \cos a + V_2 \cos \beta)$ and this is also equal to the effort P. Therefore the energy given to the wheel per lb. of steam per second is $PS = \frac{S}{g}(V_1 \cos a + V_2 \cos \beta)$.

Hence $\frac{V_1^2 - V_2^2}{2g} = \frac{S}{g}(V_1 \cos a + V_2 \cos \beta)$.

It may be noted here that $V_1 \cos a + V_2 \cos \beta = w$ may be taken directly from the velocity diagram Fig. 511, This diagram also shows that $V_1 \cos a + V_2 \cos \beta = R_1 \cos \theta_1 + R_2 \cos \theta_2$.

On entering the buckets the velocity of the steam *in the direction parallel to the axis of the wheel* is represented by AN in Fig. 510 and by N1 in Fig. 511 and this is equal to $V_1 \sin a = u_1$. On leaving the buckets the velocity of the steam in the direction parallel to the axis of the wheel is represented by *na* in Fig. 510 and by *n*2 in Fig. 511 and this is equal to $V_2 \sin \beta = u_2$. The velocities u_1 and u_2 are the velocities of flow of the steam in the direction parallel to the axis of the turbine at entrance to and exit from the buckets, respectively.

In passing through the buckets the change in the momentum of the steam per lb. per second, in the direction parallel to the axis of the wheel, is $\frac{1}{g}(V_1 \sin a - V_2 \sin \beta)$ and this is the *axial thrust* on the wheel. Since $V_1 \sin a = R_1 \sin \theta_1$ and $V_2 \sin \beta = R_2 \sin \theta_2$ the axial thrust on the wheel is also equal to $\frac{1}{g}(R_1 \sin \theta_1 - R_2 \sin \theta_2)$.

294. Effect of Friction in Impulse Buckets.—The effect of friction in the buckets of an impulse turbine is to make R_2 less than R_1. Let $R_2 = kR_1$, where k is a *coefficient of velocity* or *friction factor*. According to Professor Rateau, who made numerous and careful experiments to determine the value of k, when the buckets are very carefully made, smooth, and with sharp inlet edges, the value of k may be as high as 0·85, but it is expedient to reckon on k being from 0·75 to 0·80.

The loss of energy per lb. of steam due to friction in the buckets is $\frac{R_1^2 - R_2^2}{2g} = \frac{R_1^2(1 - k^2)}{2g}$.

The loss due to the kinetic energy in the steam leaving the buckets is $\frac{V_2^2}{2g}$ per lb. of steam.

Hence the energy given to the wheel, or the indicated work, per lb. of steam is $\frac{V_1^2 - R_1^2(1 - k^2) - V_2^2}{2g}$.

The tangential effort P per pound of steam is still equal to $\frac{1}{g}(V_1 \cos a + V_2 \cos \beta)$, but $V_2 \cos \beta$ has a smaller value when there is friction than it has when there is no friction. The tangential effort P is also equal to $\frac{1}{g}(R_1 \cos \theta_1 + R_2 \cos \theta_2) = \frac{R_1}{g}(\cos \theta_1 + k \cos \theta_2)$.

The axial thrust per lb. of steam is equal to $\frac{1}{g}(V_1 \sin a - V_2 \sin \beta)$ which is equal to $\frac{1}{g}(R_1 \sin \theta_1 - R_2 \sin \theta_2) = \frac{R_1}{g}(\sin \theta_1 - k \sin \theta_2)$. If there is no axial thrust $k \sin \theta_2 = \sin \theta_1$.

295. Efficiency of Impulse Buckets.—The efficiency of the buckets of an impulse turbine, or, as it may be called, the efficiency of the wheel, is the ratio of the energy given to the wheel to the kinetic energy in the steam as it enters the buckets. This is also called the *diagram efficiency* because the quantities involved are obtained from the velocity diagram.

$$\text{Without friction, efficiency} = \frac{V_1^2 - V_2^2}{V_1^2}$$

$$\text{With friction, efficiency} = \frac{V_1^2 - R_1^2(1 - k^2) - V_2^2}{V_1^2}$$

296. Speed of Impulse Buckets for Maximum Efficiency.—If, as is common, the buckets are symmetrical about the centre plane of the wheel, then $\theta_1 = \theta_2$, and this will be assumed in what follows. It will also be assumed that V_1 and a are given.

(1) *Without friction.*—When there is no friction the efficiency of the buckets is $\frac{V_1^2 - V_2^2}{V_1^2}$ and this will be a maximum when V_2 minimum.

Refering to Fig. 512; since there is no friction, $R_2 = R_1$ and since $\theta_2 = \theta_1$ the perpendiculars *an* and AN are equal and the position of the point *a* is independent of the bucket speed S. Hence V_2 will be a minimum when *ab* coincides with *an* as in Fig. 513. Then $\beta = 90°$, BC = CN, and $S = \frac{1}{2}V_1 \cos a$. Also $\tan \theta_2 = \tan \theta_1 = 2 \tan a$.

If $a = 20°$, then $S = 0{\cdot}47V_1$ and $\theta_1 = \theta_2 = 36°$.

(2) *With friction.*—When there is friction the efficiency of the buckets is $\frac{V_1^2 - R_1^2(1 - k^2) - V_2^2}{V_1^2}$.

Referring to Fig. 514, $R_1^2 = V_1^2 + S^2 - 2V_1S \cos \alpha$,
Also, $V_2^2 = R_2^2 + S^2 - 2R_2S \cos \theta_2$, but $R_2 = kR_1$ and
$R_1 \cos \theta_1 = V_1 \cos \alpha - S$, and $\theta_2 = \theta_1$
Therefore $V_2^2 = k^2R_1^2 + S^2 - 2kSV_1 \cos \alpha + 2kS^2$.

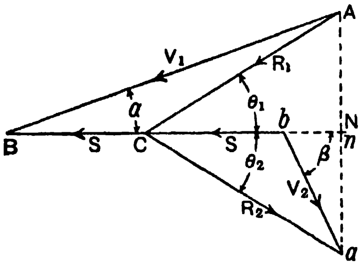


FIG. 512.

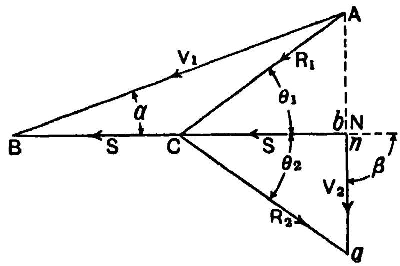


FIG. 513.

Substituting these expressions for R_1^2 and V_2^2 in the expression for the efficiency, the latter expression reduces to

$$\text{Efficiency} = \frac{2(1 + k)(V_1S \cos \alpha - S^2)}{V_1^2}.$$

This is a maximum when $V_1S \cos \alpha - S^2$ is a maximum.

Let $y = V_1S \cos \alpha - S^2$ $\quad \frac{dy}{dS} = V_1 \cos \alpha - 2S.$

And y is a maximum when $V_1 \cos \alpha - 2S = 0$, that is, when $S = \frac{1}{2}V_1 \cos \alpha$, the same result as when there is no friction. Hence, BC = CN, as shown in Fig. 515.

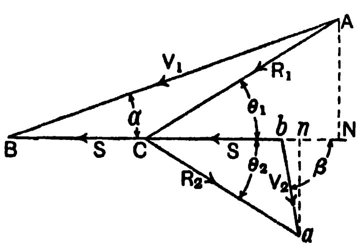


FIG. 514.

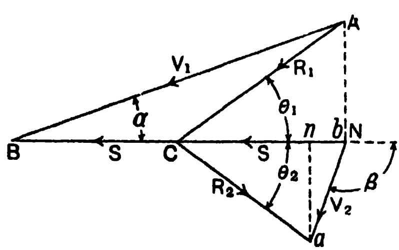


FIG. 515.

Referring to Fig. 515, $S = \frac{1}{2}V_1 \cos \alpha$, and $\tan \theta_2 = \tan \theta_1 = 2 \tan \alpha$

If $\alpha = 20°$, then $S = 0{\cdot}47V_1$, and $\theta_2 = \theta_1 = 36°$, as when there is no friction.

297. Maximum Efficiency of Impulse Buckets.—The speed of the buckets corresponding to maximum efficiency has been shown in the preceding Article to be $S = \frac{1}{2}V_1 \cos \alpha$, with and without friction.

Without friction.—Efficiency $= \dfrac{V_1^2 - V_2^2}{V_1^2}$, and when the efficiency is a maximum $V_2 = V_1 \sin \alpha$.

Therefore, maximum efficiency $= \dfrac{V_1^2 - V_1^2 \sin^2 \alpha}{V_1^2} = \cos^2 \alpha$.

With friction.—Efficiency $= \dfrac{V_1^2 - R_1^2(1 - k^2) - V_2^2}{V_1^2}$ which has been shown to be equal to $\dfrac{2(1 + k)(V_1S \cos \alpha - S^2)}{V_1^2}$.

Putting $S = \frac{1}{2}V_1 \cos \alpha$, maximum efficiency $= \dfrac{1 + k}{2} \cos^2 \alpha$.

Taking $\alpha = 20^\circ$, the maximum efficiencies corresponding to various values of k are as follows :—

Friction factor k . . .	0·7	0·8	0·9	1·0
Maximum efficiency, per cent.	75·1	79·5	83·9	88·3

298. Effect of Bucket Speed on Wheel Efficiency.—For maximum efficiency of the buckets of a simple impulse turbine it has been shown that the speed S of the buckets should be equal to $\frac{1}{2}V_1 \cos \alpha$. At this speed the bucket angle θ_1 at entrance, if there is no "shock," has been shown to be such that $\tan \theta_1 = 2 \tan \alpha$. For any other bucket speed θ_1 will have a different value if there is to be no shock at entrance, but when the buckets are made with a selected bucket angle this angle cannot of course alter and there must therefore be shock at entrance at all bucket speeds except one.

The influence of bucket speed on wheel efficiency will be best understood by considering a numerical example. It will be assumed that the angle α which the axis of the nozzle makes with the plane of the wheel is 20° and that the bucket angle at entrance is 36° which is the value of θ_1 when $S = \frac{1}{2}V_1 \cos \alpha$. Friction will be allowed for and k will be taken equal to 0·75. Also θ_2 will be taken equal to θ_1.

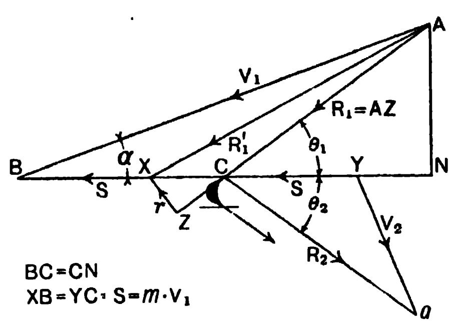


FIG. 516.

Referring to the velocity diagram Fig. 516, BC = CN = 0·47 V_1. Take XB to represent a bucket speed $S = mV_1$ (in Fig. 516 $m = 0·3$), m is the ratio of the velocity of the buckets to the absolute velocity of the steam issuing from the nozzle and is called the *speed ratio*. Then $AX = R'_1$ is the speed of the steam relative to the buckets, but this is not tangential to the buckets at entrance, the direction of the tangent at entrance being AC. Draw ZX at right angles to AC meeting AC, or AC produced, at Z. Then $AZ = R_1$, and $ZX = r$, are rectangular components of R_1' of which R_1 is tangential to the buckets at entrance and this would be the value of R_2 at exit if there were no friction. Taking $k = 0·75$, $R_2 = 0·75\ R_1$. Making $Ca = R_2$, then $Ya = V_2$.

The loss of energy in the buckets is $\dfrac{R_1^2 - R_2^2}{2g} = \dfrac{\frac{7}{16}R_1^2}{2g}$.

The loss of energy through shock may be taken as $\dfrac{r^2}{2g}$.

The loss due to the kinetic energy in the steam as it leaves the wheel is $\frac{V_2^2}{2g}$.

$$\text{Hence, efficiency} = \frac{V_1^2 - \frac{7}{16}R_1^2 - r^2 - V_2^2}{V_1^2}.$$

The values of R_1, r, and V_2 are most readily obtained from the velocity diagram drawn to scale.

For $m = 0{\cdot}3$, $R_1 = 0{\cdot}718V_1$, $r = 0{\cdot}10V_1$, and $V_2 = 0{\cdot}34V_1$ and the efficiency is 0·649 or 64·9 per cent.

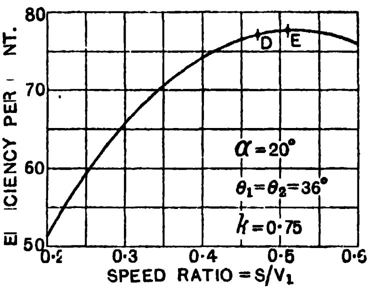


FIG. 517.

The results of tests seem to show that the shock term r^2 may be neglected. Neglecting r^2 the efficiency is 65·9 per cent. when $m = 0{\cdot}3$.

Taking different values of m and proceeding as above, neglecting r^2, and plotting the results, the curve shown in Fig. 517 is obtained. It will be found in this example that by the method just described the efficiency when $m = 0{\cdot}47$ is 77·3 per cent and is represented by the point D in Fig. 517. Also the maximum efficiency is 77·8 per cent. and is represented by the point E for which $m = 0{\cdot}52$.

299. The de Laval Turbine.—The best known simple impulse turbine is the de Laval. A pictorial view of the principal elements of this turbine is given in Fig. 518. A wheel 1 mounted on a shaft 2 has a large number of buckets 3 attached to its rim. The steam is expanded in one or more fixed convergent-divergent nozzles 4 which direct it on to the buckets on the wheel. In Fig. 518 four such nozzles are shown. The steam issues from the nozzles with a velocity between 3000 and 4000 feet per second.

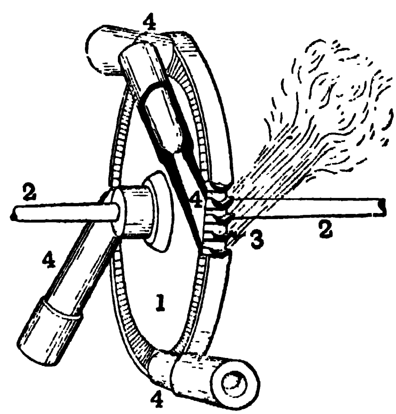


FIG. 518.

The action of the steam on the buckets has already been explained and it has been shown that for maximum efficiency the bucket speed should be about half the steam speed. In practice such a high bucket speed is not practicable. The normal speeds of some standard sizes of de Laval turbines are given in the following table.

Horse-power		5	30	100	300
Mean diameter of wheel	inches	3·94	8·86	19·69	29·92
Revolutions per minute		30,000	20,000	13,000	10,600
Mean bucket speed	ft. per sec.	515	773	1117	1384

The mean diameter of the wheel is the diameter measured to the middle of the buckets.

At the very high speed at which the wheel is run the centrifugal forces of the buckets and the body of the wheel are very great, necessitating an increasing thickness of the wheel towards its axis. The theory of the design of such a wheel is somewhat complicated and need not be considered here, but it may be mentioned that the wheel is very much weakened by a hole through its centre. For turbines up to 150 horse-power the shaft passes through the wheel, but for larger sizes the wheel is solid at the centre as shown in Fig. 519. In this design the shaft is flanged and bolted to the wheel as shown.

A point of great importance is that the slightest want of balance in the wheel would set up powerful vibrations if the shaft were quite rigid. To avoid these vibrations the shaft is made slightly flexible by having its bearings some distance from the wheel. This slight flexibility allows the wheel to run about an axis passing very nearly through the true centre of gravity of the wheel when the speed sufficiently exceeds a certain critical speed.

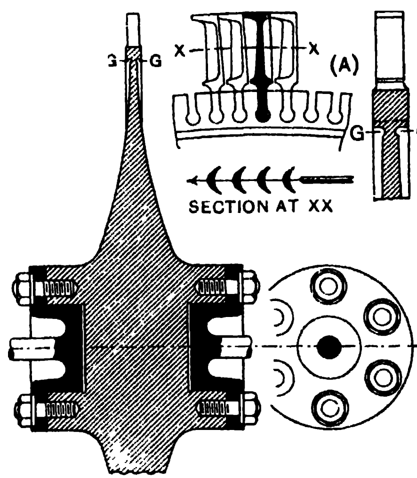


Fig. 519.

Referring to Fig. 519 it will be noticed that just under the rim the web is reduced in thickness by grooves G on the sides. The object is purposely to make this the weakest part of the wheel, so that should it give way under the centrifugal forces at excessive speed only the rim will fly off, which although serious enough would not be nearly so disastrous as the bursting of the wheel near the centre.

At (A) in Fig. 519 a portion of the rim of the wheel is shown to a larger scale, illustrating the form of the buckets and the method of attaching them to the wheel. Keyhole slots are made in the rim and the tail parts of the buckets are pressed into these slots. The outer ends of the buckets are flanged on the back and the flange of one bucket fits close against the face of the next and prevents the escape of the steam from the buckets in the radial direction. The buckets are made of steel. For the smaller sizes of wheel the buckets are stamped out, but for the larger sizes they are milled out of the solid.

In consequence of the very high speed at which the wheel of a de Laval turbine has to run it is usual to have a reducing gear as part of the engine. Two pinions on the turbine shaft gear with wheels on a second motion shaft from which the power is taken. The gear teeth are helical. To avoid end thrust on the shafts the teeth on one pinion are right-handed while those on the other are left-handed. Generally the speed reduction is 10 to 1 in the smaller turbines and 13 to 1 in the larger.

A nozzle and its casing together with the shut-off valve are shown in Fig. 520. The buckets B in the neighbourhood of the nozzle are

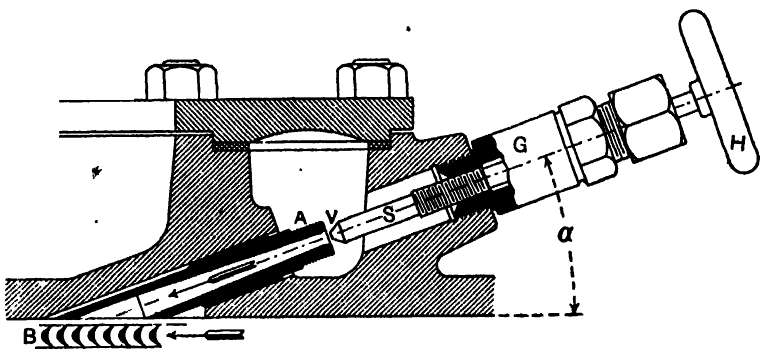


FIG. 520.

also shown. The axis of the nozzle is inclined at an angle a to the plane of the wheel; a is usually 20°. The nozzle has a straight taper. In Fig. 520 the taper ends at the right hand edge of the exit, beyond which the nozzle is cylindrical. Frequently the straight taper is continued to the end. The nozzle is forced into the hole prepared for it in the casing and it may be withdrawn by a special tool which screws on to the reduced part A after the nut and guide G for the spindle S has been removed.

The conical end V of the spindle S enters the rounded entrance to the nozzle when the spindle is advanced by turning the hand wheel H and the steam is shut off from the nozzle.

When there are several nozzles, one or more of them may be put out of action by means of the shut-off valves when the turbine is required to work at reduced loads. Governing is by throttle valve, but when the pressure is much reduced below that for which the nozzle and buckets were designed the efficiency is of course reduced.

300. Compound Impulse Turbines.—In the simple impulse turbine the steam is expanded to the full extent in one or more nozzles and the kinetic energy of the steam issuing from the nozzles is applied to drive a single ring of buckets on a wheel. It has been seen that this requires a very high bucket speed and in consequence a very high wheel speed which for many applications of the steam turbine is a disadvantage. The very high bucket speed of the simple impulse turbine also involves difficulties in construction.

There are three ways in which the wheel speed of an impulse turbine may be reduced.

(1). *Velocity compounding.*—The steam is expanded to the full extent in one set of nozzles, that is the full pressure drop takes place in these nozzles, and then passes through a ring of buckets, but it leaves these buckets with a still very high absolute velocity. The steam is then directed by means of a number of stationary buckets or guide vanes on to a second ring of moving buckets attached either to the same wheel as the first set or to a separate wheel mounted on the same shaft. The steam suffers a further reduction of absolute velocity in passing through the second set of moving buckets and may then

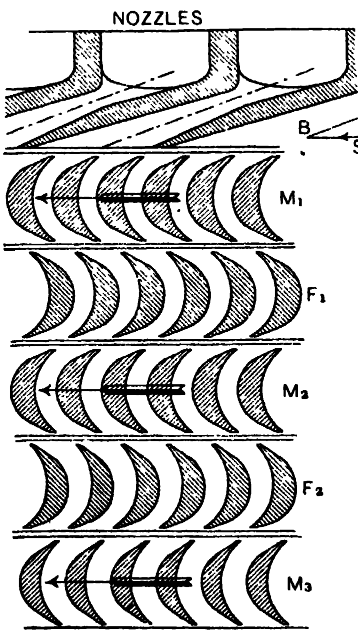


FIG. 521.

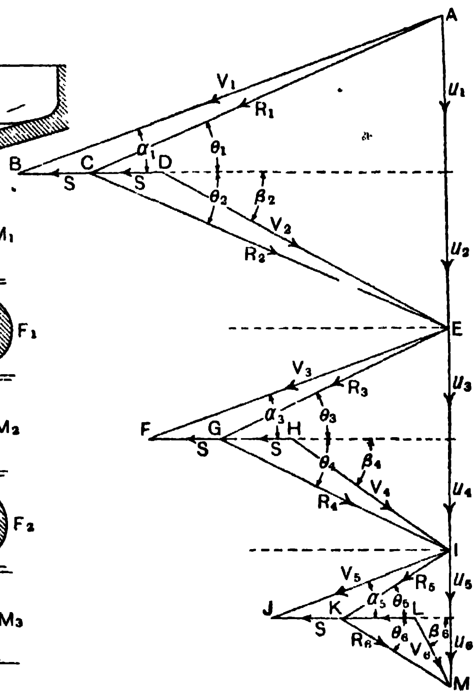


FIG. 522.

be directed by a second set of fixed buckets on to a third ring of moving buckets. The number of *velocity stages*, which is the same as the number of rings of moving buckets, is most usually two or three.

(2). *Pressure compounding.*—The pressure drop takes place in stages in separate sets of nozzles. After expanding in the first set of nozzles the steam passes through a ring of buckets carried by a wheel. After passing through these buckets the steam is further expanded in a second set of nozzles and then passes through a second ring of buckets on a second wheel mounted on the same shaft as the first wheel and so on until the full pressure drop is effected. The number of *pressure stages* is the same as the number of sets of nozzles.

(3). *Pressure-velocity compounding.*—This is combination of (1) and (2). In one or more of the pressure stages there are two or more velocity stages.

301.—Velocity-Compounded Impulse Turbines.—The arrangement of the nozzles and buckets of a velocity-compounded impulse turbine is shown in Fig. 521. There are three velocity stages in this example. M_1, M_2, and M_3 are the rings of moving buckets carried by a wheel, while F_1 and F_2 are the sets of fixed buckets attached to the casing round the wheel. The nozzles shown are in one casting.

If there are n velocity stages the bucket speed need only be about 1-nth of the speed for a simple turbine with the same initial steam speed and the same nozzle inclination.

The velocity diagrams for the different stages of the turbine illustrated by Fig. 521 are shown in Fig. 522, frictional losses being neglected. Steam leaves the nozzles at the inclination α_1 with a velocity V_1 represented by AB. CB is the bucket speed S and AC the relative velocity R_1 of the steam as it enters the first ring of moving buckets. CE = R_2 is the relative velocity and DE = V_2 is the absolute velocity of the steam as it leaves the first ring of moving buckets, DC being equal to S. θ_1 is the bucket angle at entrance and θ_2 is the bucket angle at exit for the first ring of moving buckets. In Fig. 522, $\theta_2 = \theta_1$.

The direction of DE fixes the bucket angle β_2 at entrance for the first set of fixed buckets. In the triangle EFG, EF = $V_3 = V_2$, friction being neglected, and GF = S; α_3 is the exit angle of the first set of fixed buckets and θ_3 is the bucket angle at entrance for the second ring of moving buckets; one of these two angles is chosen arbitrarily and the triangle may then be drawn. In Fig. 522, $\alpha_3 = \alpha_1$. Proceeding in the same way, the velocity diagram for the whole turbine is completed and LM = V_6 the absolute velocity of the steam as it leaves the third ring of moving buckets is obtained. In Fig. 522, $\alpha_5 = \alpha_3$, also $\theta_4 = \theta_3$ and $\theta_6 = \theta_5$.

The contracted diagram corresponding to the extended velocity diagram in Fig. 522 is shown in Fig. 523.

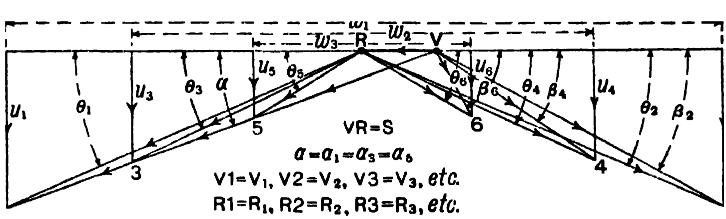


FIG. 523.

To allow for the effect of friction in the buckets, both fixed and moving, the relative velocity at exit must be reduced as in the case of the simple turbine, the velocity at exit being equal to the velocity at entrance multiplied by the friction factor k.

The driving effort on the moving buckets for the whole turbine is the sum of the driving efforts on the separate rings of moving buckets.

For a turbine with three velocity stages, the driving effort, the work done, and the axial thrust per lb. of steam per second are—

$$\text{Driving effort} = \frac{1}{g}(w_1 + w_2 + w_3).$$

$$\text{Work done} = \frac{S}{g}(w_1 + w_2 + w_3).$$

$$\text{Axial thrust} = \frac{1}{g}\{(u_1 - u_2) + (u_3 - u_4) + (u_5 - u_6)\}.$$

A numerical example for which the student should draw both the extended and contracted velocity diagrams will now be given. There are two velocity stages. The axes of the nozzles are inclined at 20° to the plane of the wheel. The velocity V_1 of the steam as it leaves the nozzles is 2500 feet per second. The bucket speed is 500 feet per second. The friction factor k is to be taken as 0·8. The exit angle θ_2 for the first ring of moving buckets is 22°, and the exit angle θ_4 for the second ring of moving buckets is 28° The exit angle α_3 for the fixed buckets is 20°.

A scale of 1 inch to 250 feet per second is suggested for the velocity diagrams. The contracted velocity diagram is shown in Fig. 524.

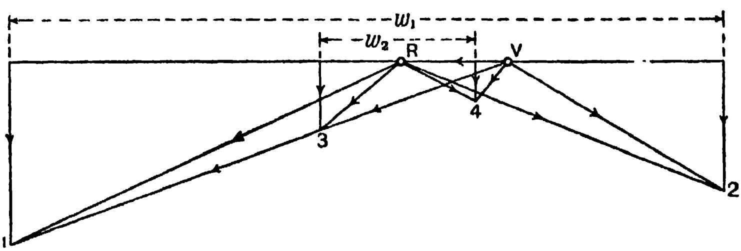


FIG. 524.

The various angles and velocities found should be taken direct from the velocity diagrams and compared with the following results found by calculation. The velocities are in feet per second, and the notation of Figs. 522 and 523 is used.

$\theta_1 = 24° \ 48'$.	$R_1 = 2038$.	$R_2 = 0{\cdot}8R_1 = 1630$.	
$\beta_2 = 31° \ 10'$.	$V_2 = 1181$.	$V_3 = 0{\cdot}8V_2 = 945$.	
$\theta_3 = 39° \ 47'$.	$R_3 = 505$.	$R_4 = 0{\cdot}8R_3 = 404$.	
$\beta_4 = 127° \ 57'$.	$V_4 = 238$.		
$u_1 = 855$.	$u_2 = 612$.	$u_3 = 323$.	$u_4 = 190$.
$w_1 = 3361$.	$w_2 = 745$.		

The following calculations may now be made, all *per lb. of steam per second*:—

$$\text{Axial thrust} = \frac{1}{g}\{(u_1 - u_2) + (u_3 - u_4)\} = 11{\cdot}7 \text{ lb.}$$

$$\text{Driving effort on first ring of moving buckets} = \frac{w_1}{g} = 104{\cdot}4 \text{ lb.}$$

$$\text{,, \quad ,, \quad second \quad ,, \quad ,, \quad ,,} \quad = \frac{w_2}{g} = 23{\cdot}1 \text{ lb.}$$

Total driving effort $= 104{\cdot}4 + 23{\cdot}1 = 127{\cdot}5$ lb.

Work done on first ring of moving buckets $\Big\} = 104{\cdot}4 \times 500 = 52{,}200$ ft.-lb.

" " second ditto, ditto $= 23{\cdot}1 \times 500 = 11{,}550$ ft.-lb.

Total work done on wheel $= 52{,}200 + 11{,}550 = 63{,}750$ ft.-lb.

The total work done on the wheel is also equal to

$$\frac{V_1^2 - (R_1^2 - R_2^2) - V_2^2}{2g} + \frac{V_3^2 - (R_3^2 - R_4^2) - V_4^2}{2g}$$

Kinetic energy in steam leaving nozzles $= \dfrac{V_1^2}{2g} = 97{,}050$ ft.-lb.

Wheel efficiency $= \frac{63750}{97050} = 0{\cdot}657$ or $65{\cdot}7$ per cent.

It will be seen that nearly 82 per cent. of all the work done on the wheel is done on the first ring of moving buckets.

302. Height Ratio in Velocity-Compounded Impulse Turbines.—Since there is no expansion of the steam as it passes through the velocity-compounded part of an impulse turbine, the volume of steam passing through each set of buckets is the same. Hence if the velocity of flow in the axial direction diminishes, which it usually does, the area through the buckets in planes normal to the axis should increase as the distance from the nozzle increases.

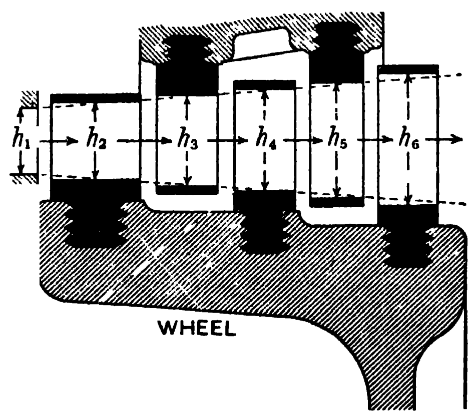


FIG. 525.

Fig. 525 shows a longitudinal section of the buckets of a velocity-compounded part of an impulse turbine, the plane of section containing the axis of the wheel. There are three velocity stages in this example. h_1 is the height of the opening of the nozzle *at exit*, and h_2, h_3, etc., are the heights of the successive sets of buckets *at exit*. If u_1, u_2, u_3, etc., are the axial velocities of the steam at the exits of the nozzle and successive sets of buckets respectively, then $\frac{h_2}{h_1} = \frac{u_1}{u_2}$, $\frac{h_3}{h_1} = \frac{u_1}{u_3}$, etc., and for the last ring of buckets $\frac{h_6}{h_1} = \frac{u_1}{u_6}$, which is called the *height ratio* for the velocity-compounded part or stage.

It is frequently assumed that the relation between the height of the buckets at exit and the distance of the bucket exit from the nozzle exit is a linear one; in that case the ends of the radial openings at exit lie on straight lines; this is not quite the case in Fig. 525, as shown by the dotted lines.

In Fig. 525 the radial openings are parallel but they are more frequently divergent as shown in Fig. 526.

If in a longitudinal section of the buckets the extremities of the openings at exit lie on straight lines, as in Fig. 526, and if the axial widths of the buckets are the same, then the heights h_1, h_2, etc., will increase by equal steps and the axial velocities u_1, u_2, etc., will decrease by equal steps.

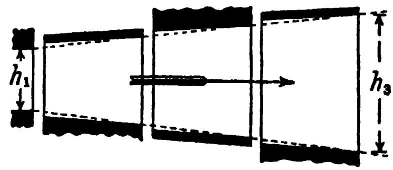


FIG. 526.

The roots of the moving buckets in Fig. 525 are let into parallel grooves turned on the wheel. The sides of these grooves have V-shaped grooves cut in them to receive teeth formed on opposite sides of the roots of the buckets. The fixed buckets are attached in the same way to a ring carried by the wheel casing. This is the design of Mr. Franco Tosi.

303. Pressure-Compounded Impulse Turbines.—The development of the pressure-compounded impulse turbine has been mainly due to the work of Professor Rateau, whose designs have been largely adopted. The designs of other inventors in this field generally follow more or less closely those of Professor Rateau, the differences being mainly in the details. The Zoelly design is well known, and while differing from the Rateau in points of detail has generally fewer stages. The Rateau turbine has sometimes as many as thirty stages while the Zoelly has seldom more than sixteen.

The first two stages and the last stage of an eight-stage Zoelly turbine are shown in Fig. 527. The space between the shaft S and the cylindrical casing C is divided into compartments by fixed cast iron diaphragms D. The forged steel wheels W, attached to the shaft S and separated at the hubs by distance rings T, rotate in these compartments. The steam nozzles N are formed in the diaphragms except the first set at the high-pressure end which are formed in the cast iron ring d. The nozzles are really annular openings subdivided by pieces of sheet nickel steel bent to the proper shape and embedded in the castings. The first set of nozzles extend round part of the circumference only but the other sets extend round the whole circumference. The buckets B are made of sheet nickel steel bent to shape. These will be further described presently.

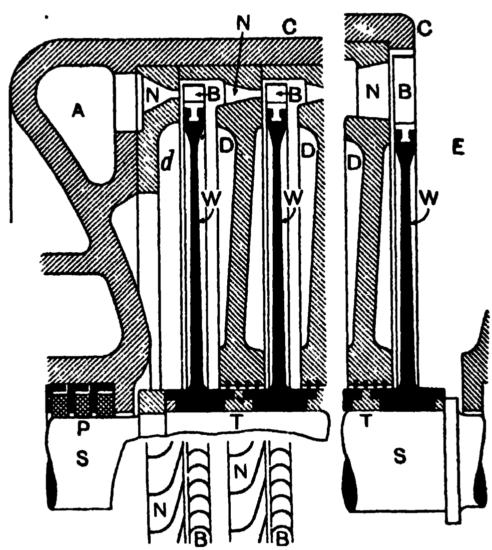


FIG. 527.

Steam is supplied to the annular steam chest A from which it passes through the first set of nozzles into the first wheel chamber. The first chamber exhausts through the second set of nozzles into the second chamber and so on to the exhaust chamber E.

The shaft S enters and leaves the turbine casing through glands having packing rings P. The turbine casing and the diaphragms are in halves, the latter being secured to the former by tap bolts. The two halves of the casing are flanged and bolted together. When the upper half of the casing is lifted off it takes with it the upper halves of the diaphragms and exposes the wheels with their buckets for inspection.

To reduce leakage of steam at the bore of the diaphragms some form of labyrinth packing is used, or there is simply a number of grooves formed in the bore.

It is important to remember that there is no expansion of the steam except in the nozzles and that the pressure on both sides of a wheel is the same. To ensure this the wheel disc has generally a number of fairly large holes in it so that there is free communication

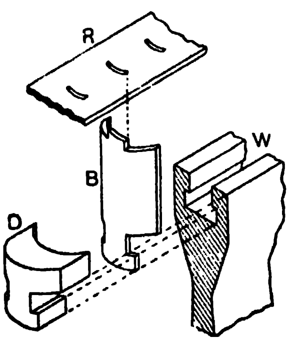


FIG. 528.

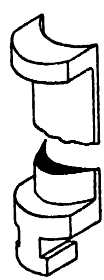


FIG. 529.

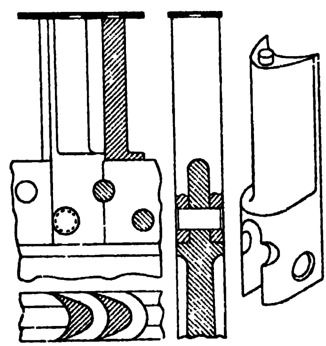


FIG. 530.

between one side of the wheel and the other for the passage of the steam. Also, the buckets being clear of the casing there is free communication over their outer ends or over the shroud ring. The motion of the steam from one set of nozzles through the buckets to the next set of nozzles is of course due to the kinetic energy in the steam.

The buckets of the Zoelly turbine are further shown in Fig. 528. The buckets or blades B are made of sheet nickel steel, as has already been stated. The correct spacing of the buckets is ensured by distance pieces D. A shroud ring R, in segments, is riveted to the outer ends of the buckets, the rivets being projections on the buckets. The buckets and distance pieces fit in a T-slot in the rim of the heel W.

Another form of bucket is shown in Fig. 529. This is fitted to the wheel in the same way as the Zoelly bucket but the bucket and distance piece are in one, the whole being machined from the solid bar. In place of a shroud ring each bucket has a flange on the back at the outer end, as in the de Laval buckets illustrated in Fig. 519, p. 417.

The Rateau design of bucket is shown in Fig. 530. These buckets straddle the rim of the wheel to which they are riveted. The buckets are made of nickel steel and are machined out of the solid bar. A shroud ring, in segments, is riveted on as shown.

304. Theory of Pressure-Compounded Turbines.—The behaviour of the steam as it passes through the successive stages of a pressure-compounded turbine is best studied by reference to the Mollier diagram or total heat-entropy chart. A definite example will be taken. It will be assumed that steam having an initial absolute pressure of 180 lb. per square inch and superheated 100° C. (180° F.) is used. This steam is supplied to a pressure-compounded impulse turbine in which there are eight stages and therefore eight wheels. The pressures in the successive chambers will be assumed to be 110, 60, 30, 16, 8, 4, 2, and 1 lb. per square inch absolute, the last pressure being that in the exhaust chamber E, Fig. 527.

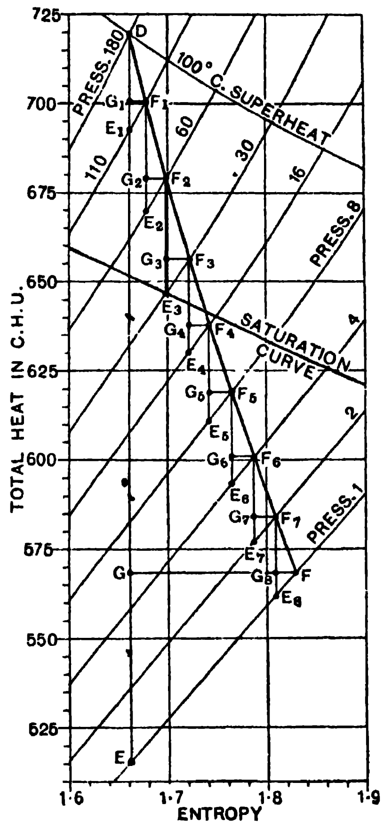


Fig. 531.

A portion of a Mollier diagram is represented in Fig. 531, but, for the sake of clearness, only those lines of the diagram are shown which are necessary for the example.

The condition of the steam as it enters the first set of nozzles is represented by the point D. The steam expands adiabatically in the first set of nozzles to a pressure of 110 lb. per square inch which becomes the pressure in the first wheel chamber. This expansion is represented by the vertical line DE_1, a line of constant entropy, lying between the constant pressure lines 180 and 110. Owing to the friction of the steam in the nozzles and buckets, and also the friction of the revolving wheel against the steam surrounding it, there is a loss of kinetic energy, apart from that due to the useful work done on the buckets, but this friction loss is converted into heat which is returned to the steam. The friction loss in the first stage of the turbine, measured in heat units is represented by E_1G_1, and the total heat in the steam as it enters the second set of nozzles is shown by the level of the point G_1. But the pressure being 110 the condition of the steam is now represented by the point F_1 where the horizontal line through G_1 cuts the constant pressure line 110.

It will be assumed that the friction loss E_1G_1 is 0·3 of DE_1, which means that the efficiency of the first stage is 0·7, and this will be assumed to be the efficiency of each of the other stages.

In the first stage DE_1 is the theoretical heat drop neglecting friction and DG_1 is the net heat drop for that stage.

In the second stage the theoretical heat drop is F_1E_2 and F_1G_2 is the net heat drop for that stage. And so on until the last stage is reached in which the theoretical heat drop is F_7E_8 and the net heat drop is F_7G_8. The final condition of the steam as it leaves the buckets of the last wheel is represented by the point F.

Had the steam been expanded from the initial pressure of 180 to the final pressure of 1 lb. per square inch, without loss, the heat drop would have been DE, which is obviously less than the sum of the theoretical heat drops in the different stages. Denoting DE by Q and the sum of the successive theoretical heat drops by q, the ratio of q to Q is called the *reheat factor* for the turbine and expresses the effect of the reheating of the steam due to friction in the different stages. The reheat factor will be denoted by R.

The efficiency of all the stages combined is DG/DE = e_i and is called the *internal efficiency* of the turbine. Assuming that each stage has the same efficiency e_s, called the *stage efficiency*, then since the net heat drop in any stage is equal to the theoretical heat drop multiplied by e_s, and since DG is the sum of the net heat drops, DG = qe_s. But $q = \mathrm{RQ}$, therefore $e_i = \dfrac{qe_s}{\mathrm{Q}} = \mathrm{R}e_s$, which shows that the efficiency of the combined stages is greater than the efficiency of a single stage, since R is greater than 1. This is a special feature of the pressure-compounded turbine due to the fact that the loss due to friction in one stage is partly recovered in the next. There is no such recovery in the velocity-compounded turbine.

Returning to the numerical example: careful measurements give the following results: DG = 151 C.H.U., DE = 204 C.H.U., therefore $e_i = \frac{151}{204} = 0{\cdot}740$. $q = 215$ C.H.U., therefore $\mathrm{R} = \frac{215}{204} = 1{\cdot}054$. But since $e_s = 0{\cdot}7$ and $e_i = \mathrm{R}e_s$, therefore $e_i = 1{\cdot}054 \times 0{\cdot}7 = 0{\cdot}738$, which is in fairly close agreement with $e_i = 0{\cdot}740$ obtained above.

The curve joining the points D, F_1, F_2, . . . F (Fig. 531) is called the *condition curve* for the turbine and is the locus of the points which represent the condition of the steam at the beginning and end of each stage. The condition curve is very flat, especially in the wet steam field where it is practically a straight line. It will be found that the condition curve for not less than six stages is practically the same as that constructed for any larger number of stages.

Having obtained the condition curve, using, say, eight stages, the pressures may be re-arranged to give, say, the same heat drop in each stage which is a common practice, and the same condition curve may be used to find suitable pressures for a larger number of stages.

The student will find it an instructive exercise to take the data of the foregoing example and draw the corresponding figure on a temperature-entropy chart. A study of the figure on the temperature-entropy chart will help greatly the understanding of the behaviour of

the steam in the pressure-compounded turbine, but for practical purposes the total heat-entropy chart should be used.

305. Pressure Compounding and Velocity Compounding Combined.—The primary object in compounding impulse turbines is to reduce the bucket speed S. For a given heat drop U and a given reduction of bucket speed this can be done in fewer stages by velocity compounding than by pressure compounding. For example, let the heat drop in a pressure stage be 30 C.H.U. The velocity of the steam issuing from the nozzles is $V = 300\sqrt{30} = 1643$ feet per second. Taking the bucket speed at one-third of V, $S = 548$. Now if into this pressure stage two velocity stages be introduced, S may then be halved, or $S = 274$. If two pressure stages be introduced, each with the same heat drop, then $V = 300\sqrt{15} = 1162$. Taking the bucket speed at one-third of this, $S = 387$. It would require four pressure stages to take the place of two velocity stages, neglecting the reheat factor. A comparison of the mechanism of two velocity stages and four pressure stages will show that the former is much simpler than the latter. It must be remembered however that pressure compounding is more efficient than velocity compounding on account of the reheating effect as was shown in the preceding Art.

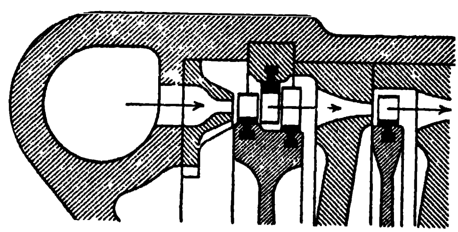


FIG. 532.

A common type of pressure-compounded impulse turbine has the first pressure stage velocity-compounded as shown in Fig. 532. Here there are two velocity stages in the first pressure stage, but there is no velocity compounding in any of the other pressure stages.

The Curtis is a well known type of pressure-compounded turbine in which all the pressure stages are velocity-compounded. An example of a Curtis turbine as used for ship propulsion is illustrated and described in the next Art.

306. Curtis Marine Turbine.—For the following particulars of an example of the Curtis marine turbine the author is mainly indebted to a translation from the Italian of a paper by Commr.-Ingr. Vittorio Mallfatti, published in *Engineering*, Jan. 31, 1913.

The turbine to be described is part of the power equipment of the Italian scouts *Marsala* and *Nino Bixio*. Each ship has three Curtis turbines, each driving a propeller. Each turbine casing has within it an ahead turbine having sixteen pressure stages and an astern turbine having two pressure stages.

Seven of the pressure stages of the ahead turbine are shown in Fig. 533. The first pressure stage has four rings of moving blades, and therefore four velocity stages; then follow five pressure stages each with three rings of moving blades, and the remaining ten have each two rings of moving blades. The total number of rings of moving blades is therefore 39.

The moving blades of the first five pressure stages are mounted

on five separate wheels while those of the remaining eleven pressure stages are fitted to a drum.

The casing C is of cast iron, divided longitudinally into a lower and an upper half, each in three parts which are connected by flanged joints, one of which is seen at F. The lower and upper halves of the ends (the upper half of the forward end is seen at H) are cast in one piece with the adjacent part of the main casing. The lower halves of the ends are bolted to the supports for the bearings of the turbine shaft. At the forward end there is a thrust bearing with four collars.

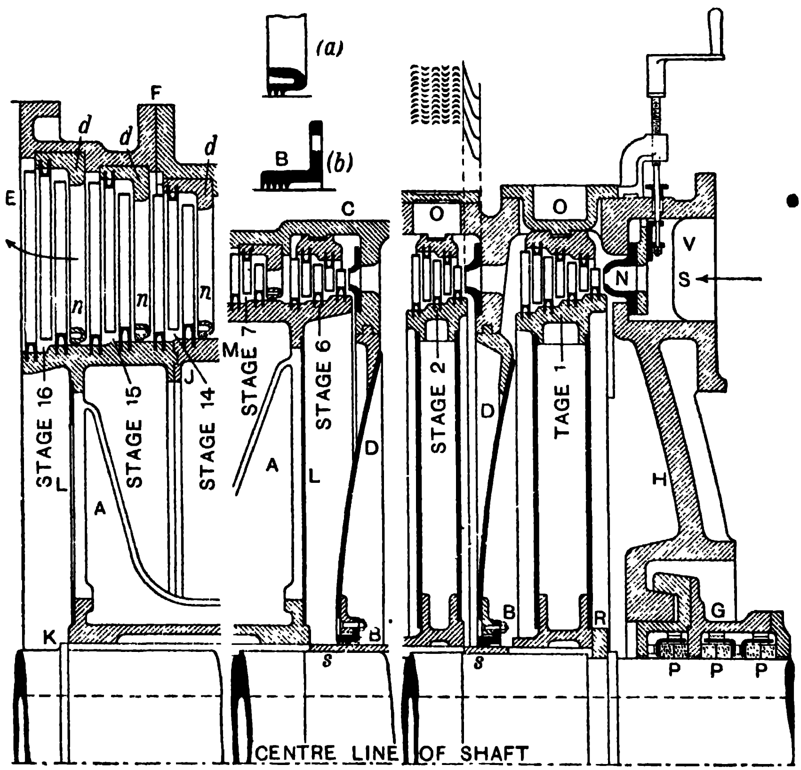


FIG. 533.—Curtis marine turbine.

The centre part of the drum is a steel casting having arms A. The rim M is of forged steel in four rings riveted together through internal flanges; one of these flanged joints is seen at J. The rim of the drum is held to the arms by end plates L riveted on to the arms and rim. The plate at the forward end entirely closes that end of the drum while the other plate has large openings in it.

The object of having a drum instead of wheels for the later stages is that the forward end of the drum being exposed to the pressure in the sixth stage while the aft end is under the pressure in the condenser there is a thrust on the shaft which opposes the thrust of the propeller.

The diaphragms D in the wheel stages are made of sheet steel, dished, and fitted as shown. At the shaft these diaphragms are connected to soft metal bushes B having internal grooves shown more clearly in detail at (*b*). The diaphragms *d* in the drum stages have similar internal grooves next to the rim of the drum as shown more clearly at (*a*). These grooves shown at (*a*) and (*b*) serve to diminish the leakage of steam, and should the diaphragms accidentally touch the drum or shaft the narrow projections present so small a surface of rubbing contact that all that happens is a slight amount of wear.

The steam inlet is at S. For the first stage there are 20 convergent-divergent nozzles N each provided with an independent shut-off valve V. The nozzles *n* in all the drum stages except the first are formed in the diaphragms *d*.

The wheels and drum are keyed to the shaft and are kept in position axially by the collar K, the ring R, and distance sleeves *s* which pass through the bushes of the diaphragms D.

The shaft, which is a steel forging bored from end to end for lightness, passes through glands in the ends of the casing. One of these glands is shown at G; they are packed with carbon rings P made in segments and are pressed radially and axially by plate springs.

Inspection doors O give access to the wheel chambers without having to raise the upper half of the casing.

In the same casing as the main turbine there is a reversing turbine (not shown in Fig. 533) which exhausts into the same exhaust chamber E. This turbine has two pressure stages, the first having five and the second four velocity stages.

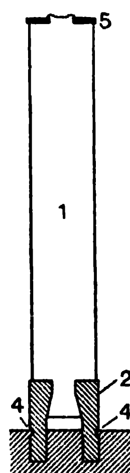


FIG. 534.

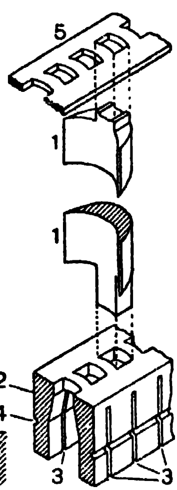


FIG. 535.

Detailed illustrations of the blading are given in Figs. 534 and 535. The blades 1 are of extruded bronze riveted to a steel foundation ring 2 which is in segments. These foundation segments are machined from solid straight bars. They have transverse saw-cuts 3 which enable them to be readily bent to the required curvature. The foundation segments are mounted in grooves in the wheel, drum, or diaphragm, and held in position by caulking the adjacent metal into the grooves 4. The shroud ring 5 is riveted on to the outer ends of the blades as shown.

At normal full power one of these turbines develops 7500 shaft horse-power at 450 revolutions per minute, and at that speed the mean blade speed is about 160 feet per second.

The particulars given in the table on page 430 are of interest.

	Stage.	Absolute pressure in front of nozzles, lb. per sq. inch.	Corresponding dryness fraction.	Absolute pressure at discharge, lb. per sq. inch.	Velocity of steam at discharge, feet per second.	Total section at throat of nozzles, sq. inches.	Arc covered by nozzles, degrees.	Mean diameter of blading, inches.
Wheels.	1	273	0·98	94	2070	--	—	80
	2	94	0·971	68	1132	21·5	69	80
	3	68	0·963	49	1132	29·7	94	80
	4	49	0·956	33	1132	40·8	130	80
	5	33	0·949	24	1132	61·7	197	80
Drum.	6	24	0·942	15·9	1132	81·7	326	80
	7	15·9	0·935	12·6	922	122·0	360	80
	8	12·6	0·929	9·8	922	154·7	360	80
	9	9·8	0·924	7·7	922	195·5	360	80
	10	7·7	0·918	6·1	922	248·9	360	80
	11	6·1	0·912	4·7	922	317·6	360	80·5
	12	4·7	0·906	3·7	922	406·1	360	80·9
	13	3·7	0·901	2·8	922	533·0	360	81·3
	14	2·8	0·895	2·2	922	679·7	360	81·6
	15	2·2	0·889	1·7	922	867·8	360	82·0
	16	1·7	0·884	1·3	922	1072·9	360	83·0

The nozzle angle is 20° in stages 1 to 14, 23° in stage 15, and 25° in stage 16.

Four out of the 20 nozzles N (Fig. 533) are reserved for use at cruising speed and are designed to expand the steam to about atmospheric pressure. The remaining 16 nozzles, which expand the steam to 94 lb. per square inch absolute, have a combined throat area of 10 square inches. Of these 16 ordinary nozzles, 12 are sufficient for the normal full power, the other 4 being reserved for overload.

The blades in the first ring of stage 1 and all the blades in stages 14, 15, and 16 have an axial width of 1 inch. All the other blades have an axial width of 0·748 inch.

Exercises XX (*b*).

1. The bucket speed of a simple impulse turbine is 1300 feet per second. The axis of the nozzle is inclined 20° to the plane of the wheel and the steam leaves the nozzle with a velocity of 3500 feet per second. Find the bucket angle at entrance if there is no shock. If the exit bucket angle is the same as the inlet angle find the absolute velocity of the steam as it leaves the buckets and the bucket efficiency, (*a*) neglecting friction, (*b*) when the friction factor is 0·75.

2. Dry saturated steam is supplied to the nozzles of a simple impulse turbine at a pressure of 200 lb. per square inch and is expanded to a pressure of 1 lb. per square inch. If the steam used per minute is 85 lb. and the shaft horse-power is equivalent to 54 per cent. of the available heat drop, find the S.H.P. and the steam consumption per S.H.P. per hour.

3. A de Laval steam turbine is provided with a nozzle placed at an angle of 20° to the plane of the wheel. Steam is supplied to it at an absolute pressure of 250 lb. per square inch, and the total heat drop is 295 B.Th.U. (163·9 C.H.U.). If 10 per cent. of this energy is lost in friction what is the velocity of the steam at discharge? The initial velocity of the steam is negligible. If the peripheral

wheel velocity is 1250 feet per second, what must be the angle of the vanes at inlet if there is to be no shock? If the relative velocity at exit is 80 per cent. of the velocity at entrance and the angle of vanes at outlet is the same as at inlet, what will be the absolute velocity of the steam as it leaves the wheel, and what will be its direction relatively to the plane of the wheel? [U.L.]

4. The inlet angle of the buckets of a simple impulse turbine is 30°. If the friction factor for the buckets is 0·75, find the outlet angle of the buckets in order that there may be no axial thrust on the wheel, assuming that there is no shock at entrance.

5. The axis of the nozzle of a simple impulse turbine is inclined at 20° to the plane of the wheel and the velocity of the steam as it leaves the nozzle is 3400 feet per second. If the inlet and outlet angles of the buckets are 30°, and the steam enters the buckets without shock, find the axial thrust on the wheel per lb. of steam per second if the friction factor for the buckets is 0·75.

6. The radius of the wheel of a de Laval turbine, measured to the middle of the buckets, is 15 inches. The speed of the wheel is 9930 revolutions per minute. The bucket angles at inlet and outlet are 36°. The axes of the nozzles are inclined at 20° to the plane of the wheel. Neglecting losses, the velocity of the steam on leaving the nozzles is computed to be 3850 feet per second. The steam consumption is 4500 lb. per hour.

Taking the loss of energy in the nozzles as 15 per cent. of the available energy, and the friction factor for the buckets as 0·75, determine:—(1) The actual velocity of the steam as it leaves the nozzles. (2) The mean speed of the buckets in feet per second. (3) The tangential effort on the wheel at the mean radius of the buckets. (4) The axial thrust on the wheel. (5) The indicated horse-power, that is, the horse-power available after deducting the friction losses in the nozzles and buckets and the kinetic energy in the steam leaving the wheel. (6) The brake horse-power, assuming a mechanical efficiency of 90 per cent. (7) The steam consumption per B.H.P. per hour.

7. Referring to Art. 298, p. 415: Assuming that the efficiency is given by the expression $\dfrac{V_1^2 - R_1^2(1 - k^2) - r^2 - V_2^2}{V_1^2}$ show that this may be put in the form $2m\{\cos a + k \cos \theta \cos (\theta - a)\} - 2m^2(1 + k \cos^2 \theta)$ and that the maximum efficiency is $\dfrac{\{\cos a + k \cos \theta \cos (\theta - a)\}^2}{2(1 + k \cos^2 \theta)}$, where $\theta = \theta_1 = \theta_2$.

Also show that if r^2 is neglected the above expressions become

$$\sin^2(\theta - a) + 2m(1 + k) \cos \theta \cos (\theta - a) - m^2\{1 + (1 + 2k) \cos^2 \theta\}$$

and $\sin^2(\theta - a) + \dfrac{\{(1 + k) \cos \theta \cos (\theta - a)\}^2}{1 + (1 + 2k) \cos^2 \theta}$

8. Draw the extended and contracted velocity diagrams for a velocity compounded impulse turbine in which there are three velocity stages, having given the following data.—Using the notation in Figs. 522 and 523, $V_1 = 2800$ ft. per sec., $S = 400$ ft. per sec., $a_1 = 20°$, $\theta_1 = \theta_2$, $\theta_3 = \theta_4$, $\theta_5 = \theta_6$, $a_3 = \beta_2$, $a_5 = \beta_4$. Assume that there is no shock at the entrances to the buckets and that there is no friction. Measure and state the values of all the bucket angles. Also compute, per lb. of steam per second, (*a*) the mean tangential effort on the wheel, (*b*) the axial thrust on the wheel, (*c*) the work done on the wheel, and (*d*) the diagram efficiency.

9. A velocity-compounded impulse wheel for a steam turbine has two rings of moving blades, and a set of fixed blades between them. The nozzle is inclined at 20° to the plane of the wheel. The exit angles of the blades are: 1st. moving, 22°; fixed, 25°; 2nd. moving, 30°. The exit height of the last blade is three times the exit height of the nozzle. The mean blade speed, which is 400 ft. per sec., is 21 per cent. of the steam velocity at exit from the nozzle.

Draw the combined velocity diagram, and find the inlet blade angles. Calculate the proportion of the heat drop converted to work on the shaft, if there is 2 per cent. disc friction loss, and the nozzle efficiency is 92 per cent. [U.L.]

10. Referring to the preceding exercise, determine the value of the friction factor *k*, (*a*) for the first ring of moving buckets, (*b*) for the fixed buckets, and, (*c*) for the second ring of moving buckets.

11. In a stage of a compound impulse steam turbine (Rateau type) the steam

is discharged from nozzles at 18° to the plane of the wheel. The mean diameter of the blade ring is 7 ft. 8 in., and the speed of rotation 800 revs. per min. The mean velocity of the blade ring is 85 per cent. of the steam velocity at exit from the nozzles. If the relative velocity at exit is 75 per cent. of that at entrance, and the absolute velocity at exit from the blades 200 ft. per sec., find (a) the blade angles at entrance and exit, (b) the work done on the blades per lb. of steam, (c) the diagram or blade efficiency, for the stage.

12. Referring to the example illustrated by Fig. 531, p. 425, and using the steam tables in the appendix, calculate the dryness fraction and the total heat for the point E, and also the heat drop DE.

13. Superheated steam is expanded in a turbine from a pressure of 180 lb. per sq. in. ab. to 1 lb. per sq. in. ab. Its initial temperature is 447° F. (230·6° C.). At a pressure of 35 lb. per sq. in. ab. it is just dry and saturated, and at 1 lb. per sq. in. ab. it is 10 per cent. wet. Sketch the temperature-entropy diagram showing the condition curve (which may be assumed straight in both superheated and saturation fields) and determine, (a) the total heat drop with isentropic expansion, (b) the actual total heat drop, (c) the percentage increase in volume of the steam at the exhaust pressure due to the reheating effect. The specific heat of the superheated steam is 0·556. [U.L.]

14. At a certain stage of a pressure-compounded steam turbine the steam enters the nozzle passages with a velocity of 270 feet per second. The heat drop in the nozzles is 17·25 B.Th.U. (9·583 C.H.U.). There is a loss of 12 per cent. of the total available energy in the nozzles, which are inclined at 20° to the plane of the wheel. The mean peripheral velocity of the wheel is 320 feet per second. Assuming that the relative exit velocity of the steam from the blades is 80 per cent. of the relative entrance velocity and that the blades are symmetrical, determine the internal or hydraulic efficiency—that is, the ratio of the work done on the blades to the heat drop in the stage. [U.L.].

15. A Curtis turbine having two pressure stages is supplied with dry saturated steam at an absolute pressure of 180 lb. per sq. in. The steam is expanded in the first stage to an absolute pressure of 18 lb. per sq. in. and further in the second stage to 1 lb. per sq. in. The stage efficiency is 0·65. With the aid of a total heat-entropy chart find, (a) the total adiabatic heat drop in the turbine, (b) the adiabatic heat drop in the first stage, (c) the adiabatic heat drop in the second stage, (d) the internal efficiency of the turbine, and (e) the reheat factor.

16. With the same bucket speed, and the same ratio of bucket speed to steam speed in a single velocity stage as in a single pressure stage, show that, to utilize the same heat drop, the number of pressure stages in a pressure-compounded impulse turbine is the square of the number of velocity stages in a velocity-compounded one multiplied by the reheat factor.

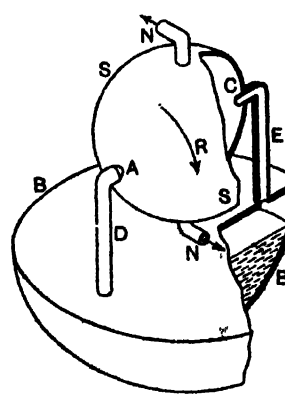


FIG. 536.

307. Pure Reaction Turbine.—There is no commercial steam turbine which works on the pure reaction principle, although, it is a curious fact, that this type of turbine was invented over twenty-one centuries ago. This ancient pure reaction turbine is generally known as Hero's engine. It was described by Hero of Alexandria about 200 B.C., but he did not claim to be the inventor. This engine is shown diagrammatically in Fig. 536. Over the boiler B there is a hollow spherical vessel S supported on bearings A and C formed on the bent ends of the supports D and E. A is a pivot bearing while C is a hollow neck bearing. Steam from the boiler passes through the hollow support E to the interior of S and issues into the atmosphere through

the nozzles N arranged as shown. The reaction of the steam against the back ends of the nozzles causes S to rotate on its bearings in the direction of the arrow R.

It is not unusual for a student to imagine at first that the reaction in a case like the above is due to the steam pushing against the atmosphere in the same way that a person propels a punt, but this is not at all like what takes place. In the reaction of the steam jet it is a question of change of momentum and the backward motion of the nozzle is analogous to the recoil of a gun.

A rare atmosphere is more favourable to the reaction than a dense one because then the steam issues with a higher velocity and there is a greater change of momentum in the steam, assuming that the boiler pressure is the same.

In any impulse turbine there is a reaction on the nozzle or on the casing containing it, but the nozzle and casing being fixed no motion due to the reaction takes place.

308. Impulse-Reaction Turbines.—In the impulse turbine all the expansion of the steam and all the heat drop take place in the stationary nozzles, and during the expansion heat energy in the steam is converted into kinetic energy, which is turned into work on the moving buckets in practically the same way that the kinetic energy of a golf club drives the ball. In the pure reaction turbine all the expansion of the steam and all the heat drop take place in the nozzles as in the impulse turbine, but the nozzles not being rigidly held are propelled backwards by reason of the reaction of the expanding steam on the nozzles, and the work done is again due to the heat drop just as in the impulse turbine. In the impulse-reaction turbine there are stationary rings of blades which form what are virtually convergent nozzles which deliver the steam on to moving blades, and the steam exerts an impulse on these moving blades exactly as in an impulse turbine, but the moving blades are of the same form as the stationary blades and like them form what are virtually convergent nozzles, so that the steam passing through them suffers a fall in pressure and a heat drop which is converted into kinetic energy just as in the pure reaction turbine. The moving blades are therefore driven by the impulse or *action* due to the kinetic energy in the steam coming from the stationary blades, and also by the *reaction* of the expanding steam between the moving blades themselves.

Turbines of the impulse-reaction type are nearly always spoken of as *reaction turbines*, but it is well to bear in mind that this is not a correct definition.

The prominent position occupied by the reaction turbine is mainly due to the inventive genius and indomitable perseverance of the Hon. Sir Charles Parsons, and this type of turbine is deservedly known as the *Parsons turbine*.

The Parsons turbine is now made by many firms of engineers the world over and, as would be expected, these engineers have made alterations and numerous improvements in regard to details and arrangements, but the main features originated by Parsons remain.

It may be mentioned here with reference to the terms *bucket* and *blade* that the former is generally used in connection with impulse

turbines and the latter in connection with reaction turbines, but there is no fixed rule in the use of these terms except that they apply to the same turbine element.

A single ring of blades in a reaction turbine will be called a *single stage*, while a ring of stationary blades together with the corresponding ring of moving blades will be called a *double stage*.

309. Action of Steam on Blades of Reaction Turbine.—The form and arrangement of the blades in two double stages of a reaction turbine are shown in Fig. 537, while Fig. 538 shows the velocity diagram. The rings of blades are very numerous and the blades are generally all alike, except towards the exhaust end, where they get flatter finishing with what are called *wing blades*, which have the form shown at (*a*) in Fig. 538. The flatter blades have a larger discharge angle and are placed at a wider pitch. The blades shown in Fig. 537 are normal blades and have a discharge angle of about 20°.

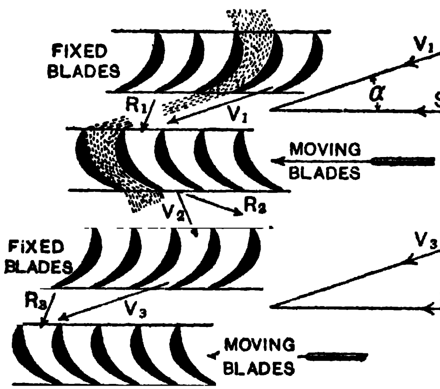


Fig. 537.

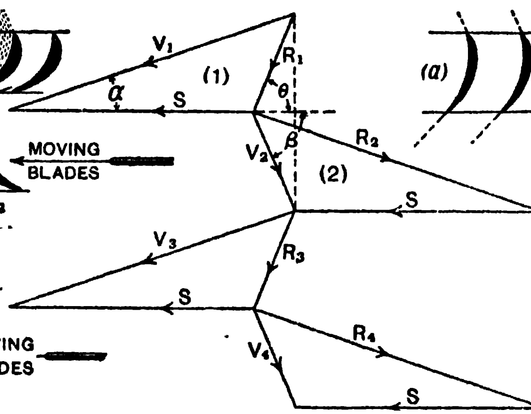


Fig. 538.

Referring to Figs. 537 and 538, the steam leaves the upper set of fixed blades with a velocity V_1, enters the adjacent ring of moving blades with a relative velocity R_1 and leaves with a relative velocity R_2 and an absolute velocity V_2. In the case of the impulse turbine R_2 would be less than R_1 on account of friction, but since in a reaction turbine there is a fall in pressure in the passages between the blades, R_2 is greater than R_1. Neglecting friction $\frac{V_1^2}{2gJ} = \frac{V_2^2}{2gJ} + U$, where J is the mechanical equivalent of heat and U is the heat drop in a single stage. Allowing for friction and leakage losses $\frac{V_1^2}{2gJ} = \frac{C_1 V_2^2}{2gJ} + C_2 U$ where C_1 and C_2 are constants. According to Mr. H. M. Martin C_1 is about 0·5 and C_2 is about 0·9. Mr. Gerald Stoney has stated that in Parsons turbines the carry-over (kinetic energy) from one row to another is about compensated for by the correction for friction, leakage, etc. This leads to the simple rule, $V_1^2 = 2gJU$ or $V_1 = 300\sqrt{U}$ where U is in C.H.U. per lb. of steam.

Since the fixed and moving blades are identical, the heat drop and

the losses may be taken as the same in the former as in the latter. It follows that the velocity triangle (1) is identical with the velocity triangle (2). Therefore $V_1 = R_2$ and $V_2 = R_1$. Also $\beta = \theta$.

The change in the velocity of whirl is $w = V_1 \cos a + V_2 \cos \beta$, but $V_2 \cos \beta = V_1 \cos a - S$, therefore $w = 2V_1 \cos a - S$.

The work done in a double stage is $\dfrac{wS}{g} = \dfrac{2SV_1 \cos a - S^2}{g}$

The energy available per double stage is $2JU = \dfrac{2V_1^2}{2g} = \dfrac{V_1^2}{g}$

The efficiency of a double stage is therefore $\dfrac{2SV_1 \cos a - S^2}{V_1^2}$ which is a maximum when $S = V_1 \cos a$ and the maximum efficiency is $\cos^2 a$. Taking $a = 20°$, the blade speed S for maximum efficiency is $0{\cdot}94V_1$ and the maximum efficiency is $0{\cdot}94^2 = 0{\cdot}88$ or 88 per cent.

310. Parsons Reaction Turbine.—The main features of a typical reaction turbine of about 5000 kw. capacity are well shown in Fig. 539. *This illustration has been prepared from a drawing which Messrs. C. A. Parsons & Co. very kindly made specially for this work.

After passing through a double-seated throttle valve, not shown, which is operated by the governor, also not shown, the steam enters the belt S and then passes through the rings of stationary and moving blades in succession to the exhaust belt E, from which it is led through the exhaust branch and exhaust expansion pipe to the surface condenser beneath the turbine.

In passing through each ring of blades the steam suffers a drop in pressure and an increase in volume, and to allow for this increase in volume and to keep the axial velocity of the steam approximately uniform the blade ring areas are increased in steps as shown.

The blade rings between one step and the next form a group or *expansion*; all the blade rings of a group have the same internal and the same external radius. The turbine shown in Fig. 539 has 38 turbine pairs, a pair consisting of one fixed and one moving ring of blades, and there would appear to be in all 12 expansions. It would, however, be more correct to say that there are 14 expansions, because in what is marked expansion 12 in Fig. 539 although the blades are all of the same height they present increasing effective areas for the passage of the steam due to the use of semi-wing and wing blades.

In the case of impulse turbines the steam pressure on the back and on the front of a moving ring of blades is the same, and it is easy to design the blades so that the axial component of the impulse on the blades shall be a negligible quantity. But in reaction turbines there is a fall in pressure at each ring of blades and a decided axial thrust when all the moving rings are considered, and this is greatly increased by the stepping of the blade rings. To balance, or partly balance, this thrust *balance pistons* or *dummies* are introduced, these being forged on or attached to the rotor.

In Fig. 539 D and *d* are dummies. Leakage of steam past the dummies is reduced to a very small amount by means of *labyrinth packing*, described and illustrated in detail in Art. 312, p. 439. Each

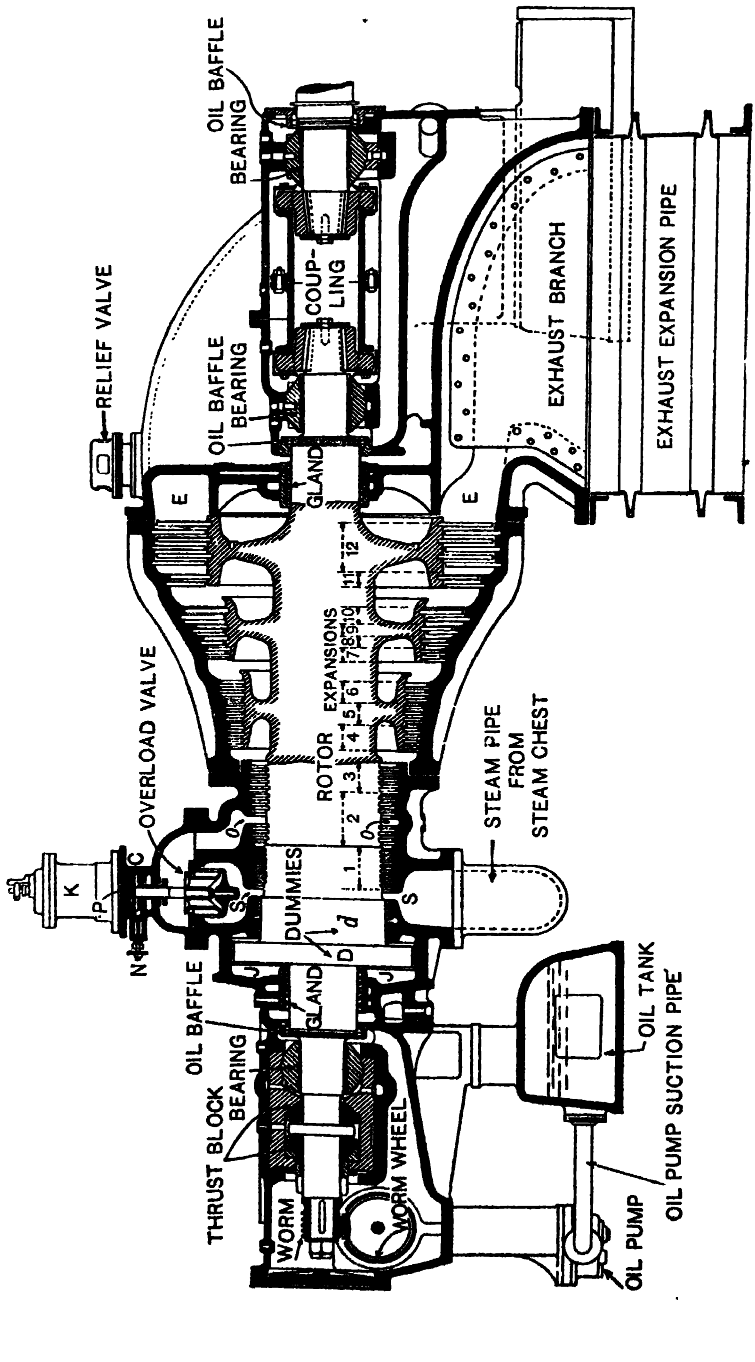


FIG. 539.—Longitudinal section through a Parsons reaction turbine.

dummy has an annular face, that of d being much smaller than that of D. The face of d is exposed to the high pressure steam in the belt S, while the face of D is exposed to the pressure of steam brought to it by a pipe, not shown, from between the third and fourth expansions. The back of D is under the pressure of steam in the space J brought there from the space between the sixth and seventh expansions by a pipe not shown. The end thrust due to the pressure drop over expansions beyond the sixth is carried by the thrust block.[1]

The rotor is a steel forging and the dummies are in one piece with it.

The overload valve is shown on the top of the steam belt S and when this valve opens steam from the belt S passes over into the belt o. The effect on the power developed is somewhat like that which would be produced in a triple-expansion engine if part of the high-pressure steam supply to the high-pressure cylinder, more or less throttled, were turned directly into the intermediate cylinder. The work done is increased but the economy in steam is slightly reduced.

The overload valve is operated as follows. Attached to the spindle of the valve there is a piston P in a cylinder C. The upper face of this piston is exposed to steam at the full stop valve pressure while the under face is exposed to steam received through the adjustable needle valve N and through a passage from the space under the throttle valve where the pressure is less than that at the stop valve. In the casing K there is a compressed helical spring which exerts an upward force on the overload valve spindle. At loads not greater than full load the resultant upward force due to the spring and the steam pressure beneath the piston P is less than that due to the steam pressure above the piston. But at a load greater than full load the governor opens the throttle valve so wide that the steam pressure beneath the piston P increases so much that the overload valve is opened.

There is forced lubrication to all the bearings. The oil pump and the governor are driven by worm gearing.

A positive thrust bearing is necessary to take the unbalanced axial thrust and to keep the rotor in correct position. The thrust bearing shown is of the Michell type, which is now rapidly superseding multiple-collar thrust bearings. For the Michell thrust block there is only one collar on the shaft which bears on blocks, each of which is a sector of an annulus and is capable of a slight tilt about a radial ridge on its back. This slight tilt is taken automatically and induces the lubricant to wedge itself, as it were, between the block and the collar.

In ordinary thrust collar bearings the permissible working pressure does not exceed 75 lb. per square inch of bearing area, but in the Michell bearing it is usually 500 lb. per square inch and may be much higher if necessary. The friction of the Michell bearing is also much less than that of the ordinary collar bearing.

The coupling between the turbine rotor shaft and the electric generator or other driven shaft is a double claw coupling which has a certain amount of flexibility and permits of an axial movement of the alternator rotor relative to the turbine spindle.

[1] In a recent Parsons marine turbine installation there are no dummies, the whole end thrust in each turbine being taken by a Michell thrust block.

The oil baffles shown prevent oil from the bearings creeping along the shaft where it is not wanted.

311. Blading of Reaction Turbines.—The blades of reaction turbines are made of brass when saturated steam is used, but copper is better for superheated steam. The blades are cut from long strips which are made of the correct cross-section by being drawn through dies.

The Parsons system of blading is shown in Fig. 540. The blades 1 have indentations 2 stamped on them at the roots. They are also thinned to a knife edge at the tips as shown at 3. Thinning the tips allows the tip clearance to be reduced. If contact should accidentally take place all that happens is a slight wear of the tips or a turning over of the knife edges. There is no danger of the blades being stripped off.

The distance pieces or spacers 4 are made of soft brass. The blades and spacers are assembled to form segments of which six or eight make a ring of blades. The blades and spacers are threaded on a wire 5 and placed in a groove in a jig which has the same curvature as the grooves into which they are afterwards to be fixed, but it is slightly

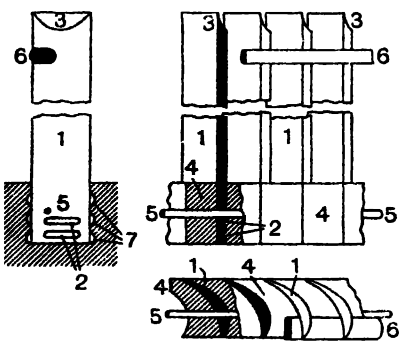


FIG. 540.

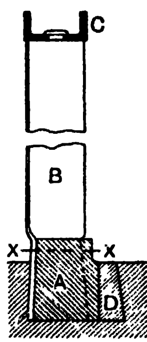


FIG. 541.

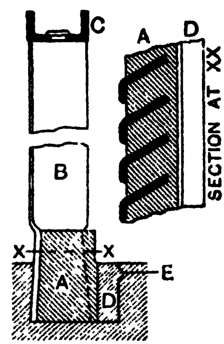


FIG. 542.

narrower. The parts are driven close together circumferentially and the ends of the wire 5 are then riveted over and brazed and then trimmed with a file. A stout wire 6 is inserted into slots in the blades near the tips. In the case of the longer blades the wire 6 is bound to the blades by thin copper wire. Finally the wire 6 and the binding wire, if any, are silver soldered to the blades. This maintains the correct pitch of the blades at the tips and strengthens the combination.

The segments of blades are placed in the grooves of the rotor or the casing, and the spacers are caulked down into the grooves and into the narrow grooves 7 on the sides of the main grooves, and also into the indentations 2 on the backs of the blades.

In the Willans and Robinson design of blading shown in Fig. 541 a segment is made up of blades B by first attaching the blades to a segmental foundation ring A. The tips of the blades are then connected by a segment of shroud ring C of the section shown. The segments of blading are placed in the grooves of the rotor or the casing. These grooves slope inwards on one side and are wide enough

to take the foundation ring A and a segmental soft brass caulking ring D.

In the Allis-Chalmers modification of the Willans and Robinson design, shown in Fig. 542, the recess for the caulking ring D is of uniform width, but there is a narrow groove E on one side as shown. In both designs the roots of the blades are stamped and then driven into saw cuts in the foundation ring as shown in the section taken at XX.

312 Labyrinth Packing.—Leakage of steam past the dummy pistons in reaction turbines is reduced to a very small quantity by the use of *labyrinth packing* which acts on the throttling principle. The steam which tends to escape is led through a succession of narrow openings which alternate with spaces in which the steam expands.

One common type of labyrinth packing is shown in Fig. 543. The casing C has a number of grooves cut in it into which are fitted brass rings R, which project into wider grooves turned on the dummy piston D. Each ring R is hollowed out on one side as shown so as to leave a narrow surface to present to the side of the groove on the dummy, which it nearly touches. This type of labyrinth packing is known as the *facial clearance* type.

The *radial clearance* type of labyrinth packing is shown in Fig.

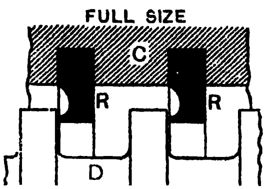


FIG. 543.

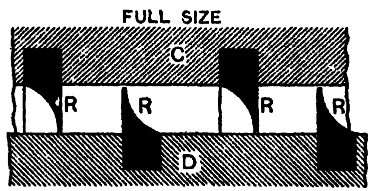


FIG. 544.

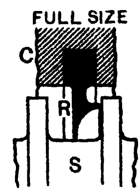


FIG 545.

544. The brass rings R in this case are placed alternately on the dummy D and casing C. This type allows of greater relative axial expansion or displacement between the dummy and the casing.

Labyrinth packing is also used in glands on the rotor shaft. Fig. 545 shows a combined facial and radial clearance type of packing as used in a gland of a Brush-Parsons turbine where C is the casing, S the shaft, and R one of the brass rings.

Where the steam next to a gland is the exhaust steam on its way to the condenser there is no tendency for the steam to escape past the gland into the atmosphere, but there would be a leakage of air inwards, and to prevent this air getting in, the gland is supplied with steam at a pressure slightly above that of the atmosphere, which expands through the labyrinth packing to the condenser.

Other designs of labyrinth packing are illustrated on p. 442 in connection with the Ljungström turbine.

313. The Ljungström Radial Flow Reaction Turbine.—In all the commercial turbines referred to in the preceding pages of this chapter the general direction of the motion of the steam is parallel to the axis of the turbine: hence these prime movers are called *axial flow turbines*. A turbine in which the working fluid travels in a radial direction is

called a *radial flow turbine*. Radial flow hydraulic turbines have long been in successful operation, but it is only since the year 1910 that the radial flow steam turbine has become a commercial success.

The Hon. Sir Charles Parsons experimented with the radial flow steam turbine, but he abandoned it in favour of the axial flow type. The brothers Ljungström, two Swedish engineers, have successfully developed a radial flow steam turbine which in many respects is a remarkable machine, the details having been worked out with great ingenuity and skill. The English makers of this turbine are the Brush Electrical Engineering Co., Loughborough.

In the Ljungström turbine there are two discs, facing one another, both carrying concentric rings of blades so placed that the rings on

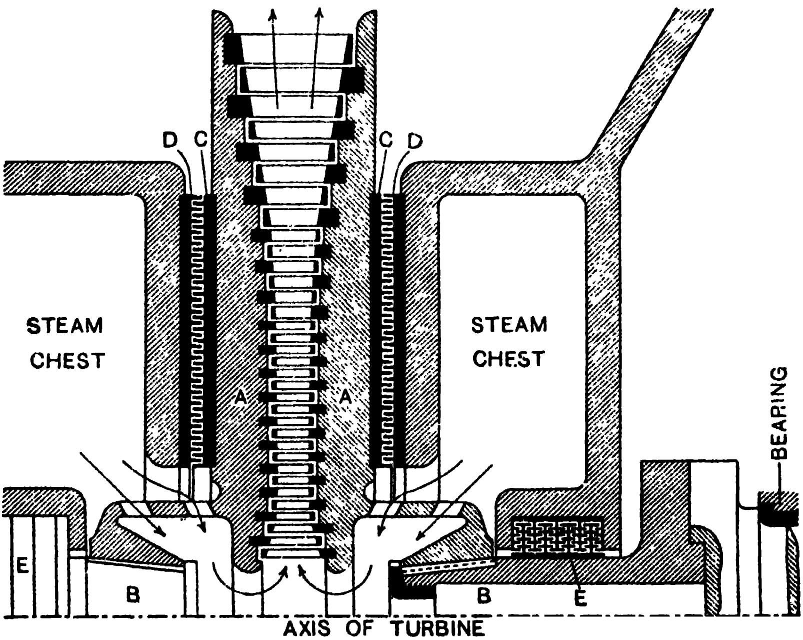


FIG. 546.—Scheme of the Ljungström turbine.

one disc interleave with the rings on the other. These alternate rings of blades correspond to the stationary and moving rings of blades of an axial flow turbine. In the Ljungström turbine of what is called the *single motion* type one disc is stationary and carries the stationary rings of blades while the other, carrying the moving rings of blades, rotates. In the *double motion* type both discs rotate at the same speed but in opposite directions. Both types work on the impulse-reaction principle and the blades are similar in form to the Parsons blades.

The principal features of a double motion Ljungström turbine are shown, more or less diagrammatically, in Fig. 546. The discs A carry the rings of blades and are mounted on the inner ends of short overhung shafts B. Steam from the steam chests passes through holes in

the hubs of the discs A and then through the rings of blades to the condenser as shown by the arrows.

The discs A carry discs C which face discs D connected to the steam chests. The adjacent faces of the discs C and D have formed on them labyrinth rings which reduce to a small amount the steam leakage to the condenser between the rotating discs and the stationary steam chests. Steam leakage into the atmosphere along the shafts is minimized by means of the labyrinth packing at E. The shafts B are coupled to the shafts of two electric generators placed one on each side of the turbine.

The actual form of the turbine discs A is shown in Fig. 547. Each disc is made of steel and is in three parts, 1, 2, and 3, which are connected by steel expansion rings 4 and 5. These expansion rings have flanges of circular cross-section which enter into grooves turned in the parts to be connected. The metal at the entrance to the grooves is afterwards closed over the flanges of the rings. The object of introducing these expansion rings is to reduce the internal stresses and prevent distortion of the parts due to differences in temperature. Fig. 547 also shows the way in which the labyrinth disc C is connected to the turbine disc A: this is by two expansion rings 6 and 7 of the type just described. The ring 7 and one flange of ring 6 are, however, first connected to seating rings which are caulked into grooves turned in the discs. The stationary labyrinth discs D (Fig. 546) are connected to the steam chests in a similar manner.

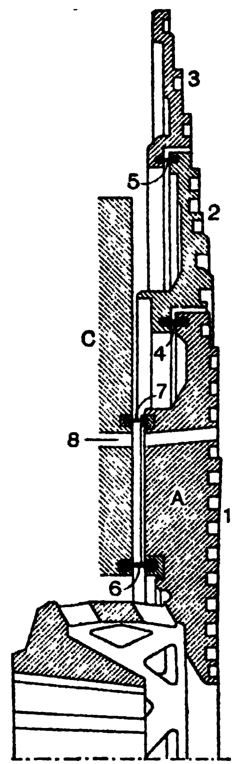


Fig. 547.

The steam pressure between each rotating labyrinth disc and its corresponding stationary disc varies from the initial pressure in the turbine to the vacuum in the condenser, just as the pressure between the blade discs varies within these limits. Advantage is taken of this fact and the labyrinth discs are so proportioned that the axial pressure of the steam, which between the blade discs tends to force them apart, is balanced in the labyrinth discs.

A ring of holes 8 (Fig. 547) form passages to allow high pressure steam to be by-passed to an intermediate stage of the blade system in order to deal with an overload.

An enlarged section of a portion of the labyrinth packing on the discs is shown in Fig. 548. The constrictions are obtained at the edges of thin nickel strips S held in grooves by caulking wire R. One of these strips in position is shown to a still larger scale at (*a*).

An enlarged section of two rings of the shaft labyrinth packing is shown in Fig. 549. One of these is a running ring and is keyed to the shaft; the other is a stationary ring and is keyed to the housing.

These rings have deep grooves turned in both sides and the intervening metal has its outer ends turned very thin, and those thin ends are then rolled over to an angle of 45°. When the rings are in position the edges of the thin bent parts make contact with the neighbouring rings as shown. In a very short time after starting the thin edges wear slightly and a very small clearance is formed which has no tendency to increase. After passing through the numerous constrictions in the packing the leakage steam is led away through a pipe, preferably to a feed-water heater where its heat is utilized.

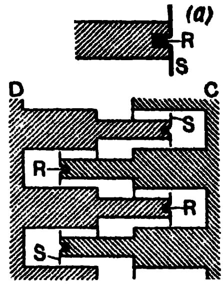


FIG. 548.

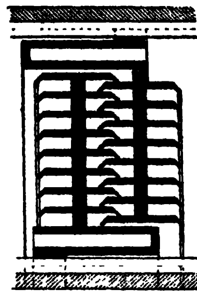


FIG. 549.

Enlarged sections of two adjacent blade rings are shown in Fig. 550. The blades 11 are machined from steel bars to the correct cross-section and cut up in the required blade lengths. By an interesting and ingenious process the rings 12 are formed and the blades welded to them. Steel strengthening rings 13 are then added. These strengthening rings have rectangular grooves turned in them which receive the dovetails of the rings 12. The metal is then cold rolled round the dovetails. Thin nickel packing strips 14, bent to U-section, are inserted into grooves in the strengthening rings and

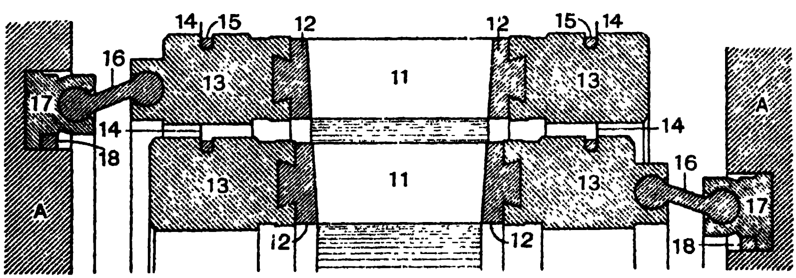


FIG. 550.

secured by caulking rings 15. The complete blade ring is attached to the expansion ring 16 which in turn is attached to the seating ring 17 which is held in a groove in the face of the disc A by a caulking ring 18.

In both types of the Ljungström turbine, since the steam enters at the centre and expands towards the circumference, the temperature of the steam surrounding the turbine is that of the exhaust and no lagging of the casing is necessary. In a natural way also, the radial flow turbine accommodates itself to the increasing volume of the steam as it flows towards the condenser; even without increasing the blade lengths the discharge area at any blade ring is proportional to its radius.

The great advantage of the double-motion turbine is that the relative velocity of an adjacent pair of blade rings is double what it

is in a single motion one or in an axial flow turbine for the same absolute blade speed. This means that for the same efficiency the axial flow turbine would require four times as many sets of blades as the double-motion radial flow turbine.

A striking feature of the Ljungström turbine is its small size for the power developed. In one of the earlier 1000 kw. machines having a speed of 3000 revolutions per minute the diameter over the outer blade ring is only about 28 inches, and the length over the flanges of the short shafts within the turbine is only about 21 inches. In this particular turbine there are 19 pairs of blade rings.

The blades are generally not more than $\frac{1}{4}$ inch wide in rather more than the first half of all the blade rings, beyond that they get wider, reaching about $\frac{3}{4}$ inch in the outer blade ring. From these facts and a study of the details it will appear that the constructing of a Ljungström turbine is more like clockmaking than ordinary engine building.

The great attention which has been given to provide for unequal expansion due to difference of temperature within the Ljungström turbine enables it to use highly superheated steam.

314. Work done on Blading of Radial Flow Turbine.—Since the blade velocity at exit in a radial flow turbine is not the same as at entrance, it is necessary to consider the change in the *moment of momentum* of the steam instead of the change in momentum only as has been done in axial flow turbines.

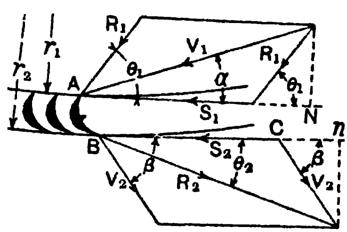


FIG. 551.

Considering first a ring of moving blades of a single-motion turbine and referring to Fig 551, let the blades have a path radius r_1 at entrance and r_2 at exit. Also, let S_1 and S_2 be the linear velocities of the blades at entrance and exit respectively.

From the velocity diagram at entrance, the velocity of whirl of the steam in the direction of S_1 is $NA = V_1 \cos \alpha$, the momentum per lb. g and the moment of this momentum about the axis of the turbine is $\frac{r_1 V_1 \cos \alpha}{g}$

From the velocity diagram at exit, the velocity of whirl in the direction of S_2 is $nC = - V_2 \cos \beta = - (R_2 \cos \theta_2 - S_2)$, the momentum is $-\frac{R_2 \cos \theta_2 - S_2}{g}$, and the moment of this momentum about the axis of the turbine is $-\frac{r_2(R_2 \cos \theta_2 - S_2)}{g}$.

The change in the moment of momentum of the steam in passing over the blade is $\frac{1}{g}(r_1 V_1 \cos \alpha + r_2 R_2 \cos \theta_2 - r_2 S_2)$, which is the torque on the turbine due to the action of the steam on the ring of blades.

If ω is the angular velocity of the rotating disc, in radians

per second, then $S_2 = r_2\omega$ and the work done per second on the ring of moving blades being considered is, $\frac{\omega}{g}(r_1 V_1 \cos \alpha + r_2 R_2 \cos \theta_2 - r_2^2\omega)$; this is in foot-pounds if the radii r_1 and r_2 are in feet and the velocities V_1 and R_2 are in feet per second.

If the speed of the rotating disc is N revolutions per minute, then, $\omega = \frac{2\pi N}{60}$.

The case of the double-motion turbine is reduced to that of the single motion one by supposing that one of the blade discs is stationary and that the other has its angular velocity doubled.

315. Combined Impulse and Reaction Turbines.—A type of steam turbine which has become common is that in which a velocity compounded turbine, such as the Curtis, is substituted for the high-pressure part of the reaction or Parsons turbine, and the whole is then called a *combined or combination turbine*. Such a turbine is also called a *disc and drum turbine* because the blades of the impulse part are mounted on a *disc* while the blades of the reaction part are mounted on a *drum*.

A certain amount of clearance must be provided at the tips of the blades of a reaction turbine and since there is a difference of pressure on the opposite sides of each ring of blades there must be a flow of steam through the clearance spaces, and this leakage of steam means loss of work. This loss of work will be greatest at the high-pressure end of the turbine because there the blades are shortest and the ratio of the clearance space to the space occupied by the blades is greatest. This objection does not apply to the velocity compounded impulse turbine because the pressure is the same all round the blades and there is therefore no inducement to the steam to flow over the tips of the blades, even when the clearance is considerable, the flow being directed through the blade channels and maintained by reason of the kinetic energy of the steam.

The combined turbine has also the advantage that it is more suitable for superheated steam than the ordinary reaction turbine. In the combined turbine the superheat is confined to the steam chest and stationary nozzles and does not reach the chamber containing the blades, which is therefore not liable to the distortion which follows the use of superheated steam in the ordinary reaction turbine.

The combined turbine has the further advantage that it is considerably shorter than the equivalent reaction turbine and the rotor is therefore shorter and stiffer. The increased stiffness of the rotor conduces to the same degree of safety with smaller clearances.

316. Relative Importance of Vacuum in Turbines and Reciprocating Engines.—The increase in the power of a turbine due to an increase in the vacuum is well shown in the temperature-entropy diagram. Taking dry saturated steam at an initial pressure of 180 lb. per square inch absolute and assuming adiabatic expansion to an absolute pressure of 2 lb. per square inch (a vacuum of about 25·9 inches of mercury) the work done in the turbine is represented by the area ABCD in Fig. 553 which will be found to be 164·3 C.H.U. or 230,020 ft.-lb. per lb. of steam.

Lowering the condenser pressure to 1 lb. per square inch (a

vacuum of about 28 inches), the work done in the turbine will be increased by the amount represented by the area DCEF which is 19·0 C.H.U. or 26,600 ft.-lb. The increase in the power due to this lowering of the condenser pressure is therefore 11·6 per cent.

Lowering the condenser pressure to 0·5 lb. per square inch (a vacuum of about 29 inches), the work done in the turbine will be greater than it was with a condenser pressure of 2 lb. per square inch by the amount represented by the area DCGK which is 36 8 C.H.U. or 51,520 ft.-lb, and the increase in the power of the turbine is 22·4 per cent.

In a reciprocating condensing engine it is not usual to expand the steam in the L.P. cylinder to a lower pressure than 7 lb. per

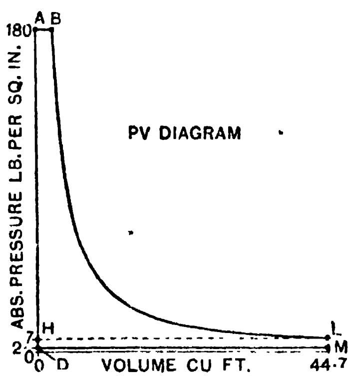


Fig. 552.

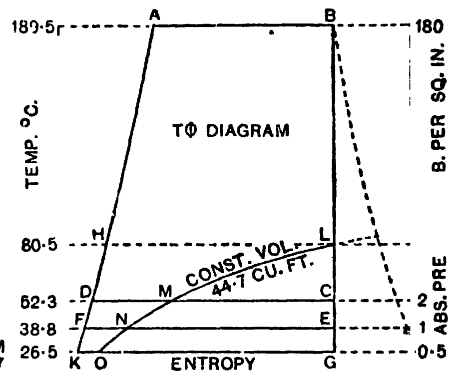


Fig. 553.

square inch absolute. The reasons for this will be given presently. Referring to the PV diagram in Fig. 552, AB represents 1 lb. of dry saturated steam at an absolute pressure of 180 lb. per square inch. The adiabatic expansion of this steam to a pressure of 7 lb. per square inch is represented by the curve BL. At the end of this expansion the volume of the steam is 44·7 cubic feet. From the point L the further expansion to a condenser pressure of 2 lb. per square inch takes place at constant volume and is represented by LM. During the whole of the return stroke the pressure is 2 lb. per square inch and this part of the cycle is represented by MD. The corresponding temperature-entropy diagram is ABLMD, Fig. 553. The area of this temperature-entropy diagram is best determined by first finding the area ABLH, using the steam tables; this is found to be 126·5 C.H.U. or 177,100 ft.-lb. The area of the part HLMD is seen from the PV diagram to be 144 × 5 × 44·7 = 32,184 ft.-lb. or 23 C.H.U. The total area ABLMD is therefore 149·5 C.H.U. or 209,284 ft.-lb.

Lowering the condenser pressure to 1 lb. per square inch, the work done in the reciprocating engine will be increased by the amount represented by the area DMNF, which is 6437 ft.-lb. or 4·6 C.H.U. This is an increase of 3·1 per cent. against the increase of 11·6 per cent. in the case of the turbine.

Lowering the condenser pressure to 0·5 lb. per square inch, the work done in the reciprocating engine will be greater than it was with a condenser pressure of 2 lb. per square inch by the amount represented by the area DMOK, which is 9655 ft. lb. or 6·9 C.H.U. This is an increase of 4·6 per cent. against the increase of 22·4 per cent. in the case of the turbine.

Before the reciprocating engine could realize the full benefit of the higher vacuum it would be necessary to expand the steam in the L.P. cylinder to the condenser pressure before release. With a condenser pressure 2 lb. per square inch the volume of the steam after adiabatic expansion is about 138 cubic feet per lb., and this would be the necessary volume of the L.P. cylinder, per lb. of steam per cycle on one side of the piston, in order to expand it to the condenser pressure. It will be seen that this volume is rather more than three times the volume of the cylinder when release takes place at 7 lb. per square inch.

The turbine must of course also be increased in size at the low pressure end to deal with the larger volume of steam resulting from the higher vacuum, but this enlargement of the turbine is not attended with friction losses as large as those introduced into the reciprocating engine by the enlarged L.P. cylinder. The enlarged cylinder of the reciprocating engine also presents a largely increased surface of metal to be alternately heated and cooled by the steam of varying temperature, while in a turbine any particular part is always at the same temperature. The uniform pure torque on the turbine shaft is also in favour of the turbine.

317. Exhaust Turbines.—The fact that the low pressure end of a steam turbine is much more efficient than the low pressure end of a reciprocating engine has led to the introduction of the *exhaust turbine* which uses the steam from reciprocating engines which are non-condensing. The exhaust turbine is chiefly used where there are a number of reciprocating engines which work intermittently, and, of necessity, are non-condensing, such as rolling mill and colliery engines. The exhaust steam from these engines, which would otherwise pass into the atmosphere and be wasted, is expanded in an exhaust turbine and then condensed.

In general it is necessary in such a plant to have a *regenerative steam accumulator* or *heat accumulator* which collects the more or less irregular supply of steam from the non-condensing reciprocating engines and delivers it to the turbine at the rate required. The regenerative steam accumlator was first introduced by Professor Rateau.

As now used the heat accumulator is a large vessel containing water into which the excess steam supply is injected and condensed, this being accompanied by a rise in pressure in the accumulator. When the supply of steam to the accumlator is less than that demanded by the turbine the pressure falls in the accumulator and part of the water in it evaporates. The action of the accumulator depends on the fluctuation of pressure within it, but this fluctuation is generally less than 2 lb. per square inch, and for a given steam supply is less the larger the accumulator.

318. Back Pressure Turbines.—Where steam is required for

heating or industrial purposes, other than power, such steam is generally of low or comparatively low pressure and may be obtained from a low-pressure boiler or through a reducing valve from a high-pressure boiler. Where power also is required it is more economical to generate high-pressure steam and use it in a turbine which instead of exhausting into a condenser exhausts at the pressure required for the steam to be used for heating or other purposes. The turbine is then called a *back pressure turbine.*

Since the demand for steam for heating or other purposes is liable to be irregular while the supply for power may be more or less constant it follows that a back pressure turbine, which has no condenser, may in many cases become very wasteful. This has led to the introduction of the *reducing turbine.*

319. Reducing Turbines.—The *reducing turbine*, or the *bleeder turbine* as it is sometimes called, is practically an ordinary condensing multi-stage turbine in which provision is made for leading away some of the steam at an intermediate stage, the steam so taken being used for heating or other purposes.

The two extreme conditions under which this type of turbine may work are: (*a*) All the steam passing through the high-pressure part leaves the turbine for heating or other purposes and the low-pressure part runs idle; (*b*) no steam is required for heating or other purposes and all the steam passes through the turbine to the condenser.

320. Mixed Pressure Turbines.—*A mixed pressure turbine* may be described as an exhaust turbine with a high-pressure part added, into which high-pressure steam may be introduced to supplement the work done in the low-pressure part by the exhaust steam from non-condensing reciprocating engines.

The two extreme conditions under which a mixed pressure turbine may work are: (*a*) The amount of exhaust steam available is sufficient for the power required of the turbine and no high-pressure steam is necessary, and the high-pressure part of the turbine then runs idle: (*b*) no exhaust steam is available and the turbine works as an ordinary high-pressure condensing turbine.

321.—Geared Turbines.—To realize the advantages of the steam turbine to the fullest extent it must be run at a high bucket or blade speed, a speed which is generally limited by the stresses due to the centrifugal forces in the revolving discs or drums which carry the blades. A high blade speed involves a high shaft speed because the radius of the disc or drum is limited by the stresses already mentioned which depend on this radius as well as on the blade speed. The result is that the highest possible speed of shaft, although not high enough to realize the theoretically most efficient turbine so far as speed is concerned, is generally very much higher than the suitable or convenient speed of a shaft for dealing with the power developed, such as the propeller shaft of a ship.

Turbine designers have overcome, to a certain extent, the objection of high turbine speed by reducing it by the method of compounding, which, however, complicates the turbine and makes it larger and heavier, and the complication, size, and weight increase as the speed is reduced by compounding.

The steam consumption lines of a 5000 kw. British Westinghouse Rateau turbine under different loads are given in Fig. 556. The results are corrected to the guarantee conditions, which were:

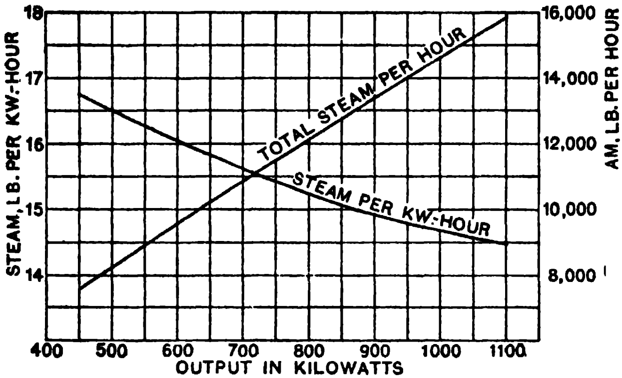


FIG. 556.

Pressure, 135 lb. per square inch by gauge. Temperature, 572·5° F. Vacuum, 28·5 inches of mercury when the barometer is 30 inches. It will be seen that the total consumption line is very approximately a Willans line.

Exercises XX (*c*).

1. In a reaction turbine the stationary and moving blades have an exit angle of 20°. The mean blade speed is 140 feet per second and the relative velocity of the steam as it leaves the blades, moving as well as stationary, is 300 feet per second. Neglecting friction and other losses, and also the small change in the volume of the steam, find the heat drop per lb. of steam in one double stage.

2. In a Parsons turbine using dry saturated steam you are given the following:—Blade angle at exit, 20° with a line transverse to the axis of the turbine; speed of steam as it issues from either guide or rotor blades, 240 feet per second; mean speed of rotor blades, 96 feet per second. Neglecting friction and other losses, find (*a*) inlet angle of acting face of blades, (*b*) British thermal units per pound of steam absorbed per row of blades, assuming the steam to act as an inexpansive fluid while passing one row. [U.L.]

3. The blade discs of a double-motion Ljungström turbine each have a speed of 3000 revolutions per minute. What is the absolute and what is the relative blade speed in feet per second at a radius of 10 inches?

4. In a group of two pairs of blade rings in a double-motion Ljungström turbine the radial pitch of the blades is 0·4 inch. The external radius of the inner ring of the group is 9 inches. Each rotor has a speed of 3000 revolutions per minute. All the blades of the group have an exit angle of 20°. Assuming that the speed ratio $\left(\frac{S_1}{V_1} \text{ or } \frac{S_2}{R_2}, \text{Art. 314}\right)$ is 0·7 throughout the group, determine the indicated work done on the group per lb. of steam.

5. In a geared turbine the gear-wheel has 229 teeth and a pitch line diameter of 77·83 inches. The pinion has 23 teeth and runs at 3000 revolutions per minute. The useful width of the wheel and pinion is 36 inches. The horse-power transmitted is 2200. Calculate: (*a*) The pitch line speed in feet per second; (*b*) the pressure on the teeth, or driving force at pitch line, per inch of useful width of wheel; (*c*) the circumferential pitch of the teeth; (*d*) the pitch line diameter of the pinion.

6. Dry saturated steam at a pressure of 200 lb. per square inch absolute expands adiabatically: (*a*) in a turbine to the condenser pressure, which is 2 lb. per square inch absolute, (*b*) in a condensing reciprocating engine to a pressure of 8 lb per square inch absolute at the end of the stroke, the condenser pressure being 2 lb. per square inch absolute. Find in both cases the percentage increase in indicated work due to lowering the condenser pressure to 1 lb. per square inch.

7. Find the efficiency of an electric generator driven by a steam turbine, having given: Steam consumption per kilowatt hour, 12·2 lb., and steam consumption per shaft horse-power hour, 8 6 lb.

8. A steam turbine driving an electric generator has a steam consumption of 12 lb. per kilowatt hour. If the efficiency of the generator is 93 per cent. what is the steam consumption per shaft horse-power per hour?

9. Tests of a steam turbine driving an electric generator gave a total steam consumption of 37,000 lb. per hour when the output of the generator was 2,500 kilowatts, and 67,000 lb. when the output was 5000 kilowatts (full load). Find the equation to the Willans line, and calculate the probable steam consumption, in lb. per kilowatt hour, when the output is 4,000 kilowatts.

10. A full load test of a steam turbine coupled to an electric generator gave a steam consumption of 11·9 lb. per kw. hour. The total adiabatic heat drop was 230 C.H.U. (414 B.Th.U.) per lb. Compute the overall efficiency of the turbine and generator.

11. A 10,000 kw. set driven by a steam turbine is installed in a power station. Steam is supplied at a pressure of 180 lb. per sq. inch abs. and superheated 150° F. (83·3° C.). The pressure in the condenser is 1·5 lb. per sq. inch abs. The efficiency ratio of the set, *i.e.* the ratio of the output at the generator to the theoretical work available from the steam, is 0·64. Determine the steam consumption in lb. per kw. hour of the set. If the consumption on a trial were 11·1 lb. per shaft horse-power hour, find the efficiency of the generator.

[U.L.]

12. Referring to the preceding exercise, determine the efficiency ratio of the turbine.

13. A steam turbine runs, on test, under conditions of pressure, superheat, and vacuum, not specified in the guarantee. Show how the corrected consumption and the efficiency ratio may be calculated and apply the method to the following case

Guarantee figures: Pressure, 175 lb. per sq. in. by gauge; superheat, 115° C.; vacuum, 28·5 inches; barometer, 30 inches of mercury.

Actual test figures: Pressure, 127 lb. per sq. in. by gauge; superheat, 37° C.; vacuum, 28·4 inches; barometer, 30 inches of mercury.

The consumption was 16 lb. per kw. hour at the generator. Calculate the corrected consumption at guarantee conditions, and the efficiency ratio if the generator efficiency is 95 per cent

Take the following correction figures and use the total heat-entropy chart supplied. 1·40 per cent. decrease of consumption per 10 per cent. increase of pressure. 14·40 per cent decrease of consumption per 100° C. increase of temperature. 7·00 per cent. decrease of consumption per 1 inch increase of vacuum.

[U.L.]

CHAPTER XXI

CONDENSERS AND AIR PUMPS

323. Object of Condensing Exhaust Steam.—In any heat engine the amount of work done per lb. of working fluid depends on the range of temperature of that fluid in the engine, being greater the greater the range. Stated in another way, the work done is greater the greater the amount of expansion of the fluid. In any heat engine it is a more or less simple matter to expand the working fluid to a small absolute pressure, but the difficulty is to get rid of the fluid after it has expanded to less than atmospheric pressure. In internal combustion engines the working fluid cannot be used over again, and it has therefore to be discharged into the atmosphere, and the pressure of the atmosphere therefore fixes the lower limit of expansion. When steam is the working fluid, however, it may be returned to the boiler and used over again and this can be done most conveniently and most economically by first condensing it and then pumping the resulting water into the boiler.

It is not the condensation of the steam in a condenser which causes the low pressure in it; the pressure of the steam as it enters the condenser should, for greatest thermodynamic efficiency, be the same as that in the condenser, and condensing it need not lower its pressure or temperature. But by condensing the steam it may then be more economically removed to make way for more steam from the engine. The heat taken from the steam in condensing it is generally entirely lost.

The primary object of condensing exhaust steam is therefore to make it possible to remove it economically at a pressure less than that of the atmosphere after it has done its work in the engine, and therefore to enable the steam to expand to a greater extent and do more work. A secondary object is the obtaining of hot feed-water for the boiler.

The amount of heat available for conversion into work, with different back pressures and different initial pressures, is given in the following table. The amounts of heat given are the heat drops due to adiabatic expansions. The specific volumes of the steam at the back pressures are also given. The steam is assumed to be initially dry and saturated, but it will of course be wet after expansion.

Abs. initial pressure, lb. per sq. in.	Abs. back pressure, lb. per sq. in.	Vacuum,* inches of mercury.	Available heat per lb. of steam.		Specific vol. at back pressure, cubic feet.
			C.H.U.	B.Th.U.	
200	15	0·0	105	189	22·5
200	2	25·93	168	303	137
200	1	27·96	187	336	256
100	15	0·0	77	139	23·5
100	2	25·93	143	257	142
100	1	27·96	162	292	266
15	2	25·93	73	131	157
15	1	27·96	150	270	293

* Referred to 30-inch barometer.

It will be seen from the table that the advantage due to lowering the back pressure, made possible by condensing the exhaust steam, is relatively greater the lower the initial pressure. Also the gain per lb. per square inch reduction in back pressure is greater the lower the back pressure.

Although the thermodynamic efficiency is greater the lower the back pressure, the ultimate efficiency may be reduced when the back pressure is reduced beyond a limit depending on the cost of producing and utilizing that back pressure. In reciprocating engines there is generally no saving by increasing the vacuum beyond 27 inches of mercury and 26 inches is often taken as the limit, but in steam turbines the vacuum may with advantage be 28 inches and in special cases it may exceed 29 inches.

324. Vacuum Measurement.—The measure of a vacuum in a vessel is the difference between the atmospheric pressure and the absolute pressure in the vessel. To make this measure definite it must be converted to correspond with a standard atmospheric pressure. The standard atmospheric pressure generally taken in connection with vacuum measurement is the barometric pressure of 30 inches of mercury which corresponds to 14·74 lb. per square inch.

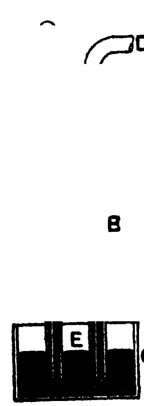


Fig. 557

The conversion of a vacuum gauge reading to standard will be best understood by referring to Fig. 557. A is a mercury barometer and B a mercury vacuum gauge, both dipping into an open basin of mercury C. The vacuum gauge is connected at D to the condenser or other vessel, the vacuum in which is to be measured. The column of mercury EF in A balances the pressure of the atmosphere acting on the surface of the mercury in C. The column of mercury EG in B plus the absolute pressure p in the vessel connected to B

at D acting on the top of the column EG also balances the pressure of the atmosphere, therefore the height GF measures the absolute pressure p above the mercury in B. But the pressure p is independent of the pressure of the atmosphere. Hence, if the reading of the vacuum gauge, corrected to standard, is h

$$30 - h = EF - EG \text{ or } h = 30 - EF + EG.$$

For example, if the gauge shows a vacuum of 27·1 inches when the barometer shows 29·3 inches the vacuum corrected to standard is $30 - 29{\cdot}3 + 27{\cdot}1 = 27{\cdot}8$ inches.

The vacuum gauge reading, corrected to standard is sometimes expressed as a percentage of the standard barometric height, thus, in the above example 27·8 inches is 92·7 per cent. of 30 inches. A vacuum so expressed must not, however, be confused with vacuum efficiency as defined in Art. 332, p. 461.

The vacuum gauges used in practice are similar to pressure gauges and are generally of the Bourdon type (see Figs. 167 and 168, p. 160), but the dials are graduated to read inches of mercury.

It may be noted here that 1 inch of mercury corresponds to 0·491 lb. per square inch, and 1 lb. per square inch corresponds to 2·036 inches of mercury.

For rough calculations 1 lb. per square inch may be taken as equivalent to 2 inches of mercury.

325. Elements of a Condensing Plant.—For the purpose of maintaining a vacuum by the condensation and removal of the exhaust steam, the principal requirements are:—

(1) A condenser in which the steam is condensed.

(2) A supply of injection or cooling water.

(3) A pump to circulate the cooling water when a surface condenser is used.

(4) A pump, called the *air pump*, for removing the condensed steam and the air and uncondensed water vapour from the condenser. [Separate pumps may be used for (a) the condensed steam and (b) the air and uncondensed water vapour.]

(5) A receptacle for the condensed steam discharged by the air pump, called the *hot-well*, from which the boiler feed is taken.

(6) Arrangements for recooling the cooling water of a surface condenser or the excess injection water not required for boiler feed, in cases where the water has to be used over and over again for cooling or injection purposes.

326. Types of Condensers.—There are two main classes of condensers. (1) *Jet condensers* in which the water for the condensation of the steam mixes with the steam, and (2) *Surface condensers*, in which the steam is condensed by contact with metal tubes kept comparatively cool by water which does not mix with the steam.

The first class may be subdivided into—

(*a*) *Parallel flow condensers*, in which the jet or spray of water and the steam enter the condenser at the top and fall together to the bottom.

(*b*) *Counter current condensers* in which the steam flows upwards through the condenser, meeting the water which streams downwards

from the top. The air is removed at the top and the water, separately, at the bottom.

(c) *Ejector condensers* in which the steam and water mix in a series of combining cones and the kinetic energy of the steam is utilized to assist in moving the water through the condenser into the hot-well against the pressure of the atmosphere.

(d) *Barometric condensers* in which a jet condenser at a high level is provided with a long vertical discharge pipe delivering the hot water into a sump at the bottom without the aid of a pump, but an air pump is required to remove the air at the top.

The second class may be subdivided into—

(i) *Surface condensers* in which the cooling water flows through tubes while the steam flows around them.

(ii) *Surface condensers* in which the steam flows through the tubes while the cooling water flows around them.

(iii) *Evaporative condensers* in which the steam flows through the tubes which are kept comparatively cool by water trickling over them, the cooling water being evaporated and taking its latent heat of vaporization from the tubes.

Jet condensers are only used where the supply of condensing water is reasonably pure because it goes with the condensed steam to the hot-well from which the boiler feed water is taken.

In modern condensing plants the most common types of condensers are, the counter current jet condenser, and the surface condenser in which the cooling water flows through the tubes and the steam flows around them.

327. Parallel Flow Jet Condensers.—Fig. 558 illustrates a design of parallel flow condenser, together with its air pump, taken from one of Mr. Michael Longridge's reports to the British Engine, Boiler and Electrical Insurance Company. This form of condenser has been much used for reciprocating mill engines. The air pump shown is a modified Edwards pump (see p. 465). The exhaust steam from the low-pressure cylinder enters the condenser at S and meets the spray of injection water W coming from the perforated pipe P.

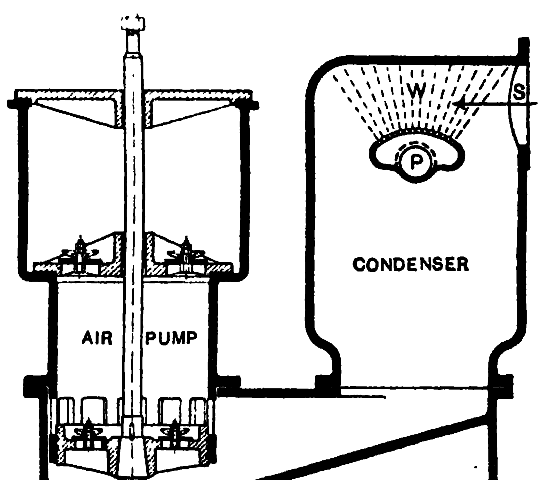


Fig. 558.

The condensate falls to the bottom of the condenser and flows to the air pump which removes it, together with the air, to the hot-well.

328. Counter Current Jet Condensers.—A vertical section through a Mirrlees-Watson counter current jet condenser is shown in Fig. 559. The injection water enters the top compartment 1 of the condenser and falls in a large number of jets through perforations in the conical shaped plate 2. The falling water is caught in the tray 3, from which it escapes in a second series of jets. The water jets from the bottom of the tray 3 descend through the short pipe 4 to the bottom of the condenser while the others are caught in the annular tray 5. The

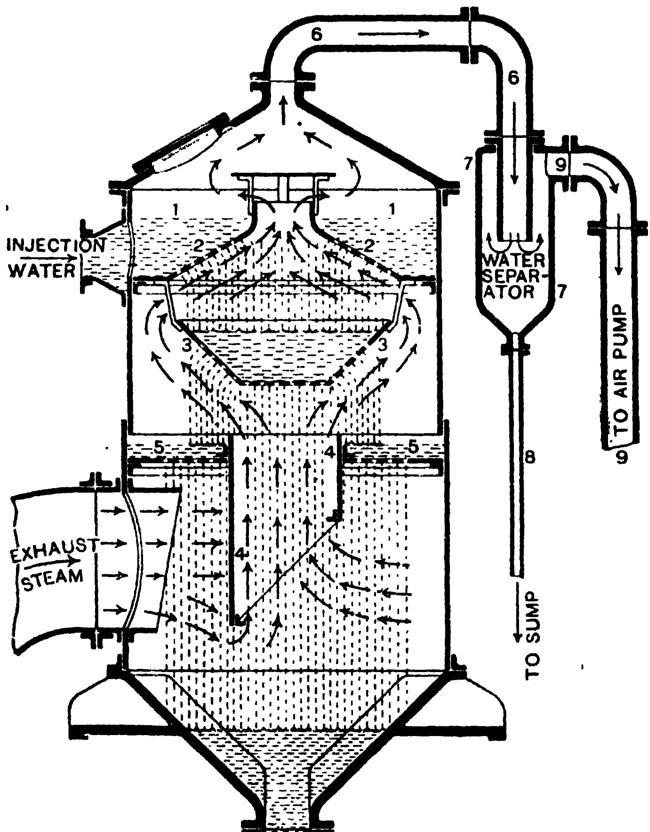


FIG. 559.—Counter current jet condenser.

water in the tray 5 escapes in a third series of jets and falls to the bottom of the condenser.

The exhaust steam and any air mixed with it enter the bottom compartment of the condenser and flow towards the top as shown by the arrows. Encountering jets of water all the way the steam is very nearly all condensed. The air and uncondensed water vapour are led away from the top of the condenser by the pipe 6 to the separator 7 where any suspended water drops out and falls through the pipe 8 to a

sump below. From the separator a pipe 9 leads the moist air to the air pump.

If the condenser is a low-level one, that is, one placed near the level of the engine, the water is taken from the condenser or from a closed collecting tank below it by a pump, generally of the centrifugal type.

If the condenser is a high-level or barometric one, that is, one the bottom of which is not less than, say, 35 feet above the level of the water in the collecting tank or sump, the condenser is self discharging so far as the water is concerned, but a pump is generally necessary to force the injection water up to the condenser.

Whether the condenser is at a low or high level the air pump has only to deal with the moist air.

329. Ejector Condensers.—Ledward's well known *ejector condenser*

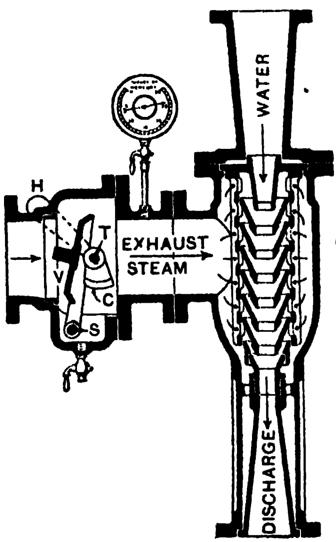


FIG. 560.

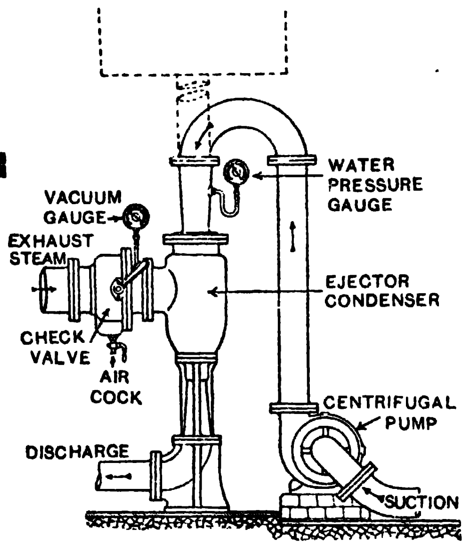


FIG. 561.

is shown in Fig. 560. The water enters at the top and flows through a number of co-axial nozzles fixed in a tube in which there are steam ports leading into the spaces between the nozzles. The water enters under a head of from 15 to 20 feet, and, in rushing across the gaps between the central parts of the nozzles, drags in the exhaust steam and air. The steam becomes condensed and the air is carried forward with the water. The ejector condenser therefore acts as an air pump as well as a condenser.

It is usual to fit a check valve V in the exhaust pipe near to the condenser. This is a non-return valve hinged on a spindle S. The use of the check valve is to prevent the backward rush of water from the discharge tank to the engine which would follow a sudden failure of the water supply to the condenser. Such a backward rush of water would close the check valve. An addition to the check valve is a

cam C attached to a spindle T operated by a handle H by which means the check valve may be locked in its closed position.

When there is no natural head for the supply of water to the condenser the arrangement shown in Fig. 561 may be used. A centrifugal pump delivers the water direct to the condenser at the necessary pressure. The dotted lines at the top of Fig. 561 show the connection of the condenser to an overhead tank when a natural head of water is available.

The ejector condenser is small in comparison with other types of condensers and there being no air pump the first cost is low. It is also simple and reliable, but it is generally only used for comparatively small engines. The vacuum obtained is generally about 24 inches. The vacuum may however be raised by increasing the water supply, but it is not found to be economical to use this type of condenser where a high vacuum is required.

330. Surface Condensers.—The chief features of the most common type of surface condenser are:—(1) A shell, placed horizontally, and having a cross-section which is either circular or narrows towards the bottom. (2) Flat tube plates bolted on to and covering the ends of the shell. (3) Water boxes open to and surrounding the tube plates. The water box at one end has a horizontal partition about half-way up, dividing it into lower and upper compartments A and B. The other box has only one compartment C. (4) A large number of tubes extending between the tube plates and providing communication between the water boxes. (5) A large opening D in the top of the shell, preferably rectangular in shape. (6) A smaller opening E in the bottom of the shell.

The operation of this condenser is as follows:—Cooling water is pumped into the water compartment A from which it passes through the lower nest of tubes to the water compartment C, returning through the upper nest of tubes to the compartment B, which it leaves through an outlet at the top. The steam enters the shell by the opening D and passing over the tubes is condensed and leaves by the opening E and passes to the air pump.

It is generally agreed that the steam should flow as directly as possible across the tubes and that baffle plates which cause a sudden change in the direction of flow should be avoided, but directing plates may be used to ensure that all the tubes throughout their length shall be as fully effective as possible.

Figs. 562 and 563 show the "uniflux" surface condenser designed and made by Messrs G. & J. Weir. Fig. 562 is a vertical longitudinal section while Fig. 563 is a vertical cross-section through the inlet and outlet water compartments. The shell S is built up of steel plates. The water boxes are of cast iron and they are provided with inspection doors O. The exhaust steam enters the condenser at D and the air and condensed steam leave at E. Two intermediate transverse plates J, through which the tubes pass, serve to prevent the deflection of the tubes and keep them at the correct pitch throughout their length; these plates also serve to direct the steam in its passage over the tubes.

The cooling water enters the chamber A at F and passes through the lower nest of tubes to the chamber C and then returns through

the upper nest of tubes to the chamber B from which it leaves by the outlet G.

In order to keep the velocity of the steam approximately uniform as it condenses, the cross-section of the body of the condenser is gradually reduced in width towards the bottom as shown in Fig. 563. For the same reason the tubes in the lower nest are generally placed

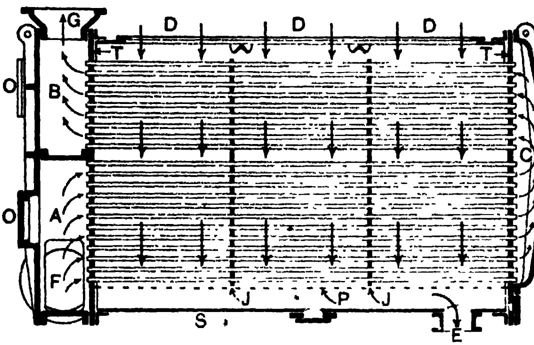


FIG. 562.

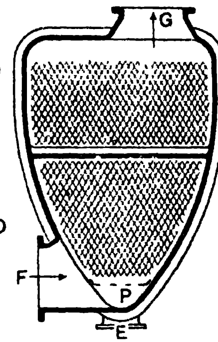


FIG. 563.

closer together. A perforated baffle plate P is fixed just under the bottom nest of tubes. The tube plates are supported by bar stays, not shown, which extend from one tube plate to the other.

The Westinghouse design of surface condenser is shown in Figs. 564 and 565, prepared from illustrations in a paper by Mr. A. E. Leigh Scanes in the *Proceedings of the Institution of Mechanical Engineers* for 1913. The important difference between this condenser and the one

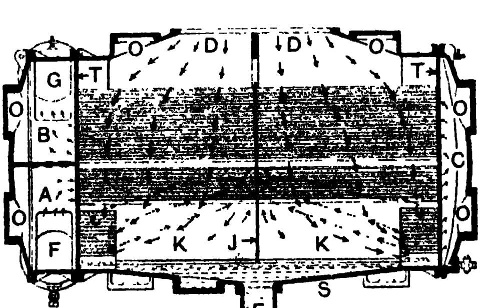


FIG. 564.

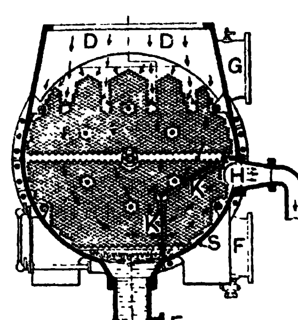


FIG. 565.

shown in Figs. 562 and 563 is that the air and condensed steam are taken separately from different parts of the condenser. The condensed steam outlet is at E in the bottom of the condenser while the air outlet is at H on the side of the condenser. A baffle plate K facing the air outlet serves to prevent the steam following the air which passes round the ends of the baffle plate. The longitudinal section (Fig. 564) is partly through the centre and partly at the vertical part

of the baffle plate K which explains the apparent interruption in the tubes towards the bottom; all the tubes extend from one tube plate to the other.

In the upper part of the upper nest of tubes it will be seen from Fig. 565 that there are gaps which serve to give greater area for the flow of the steam where there is most steam; this conduces to more uniform velocity of flow.

The shell S is of cast iron in the example shown in Figs. 564 and 565. The bar stays for the tube plates will be seen amongst the tubes in Fig. 565.

The two condensers illustrated in Figs. 562 to 565 are called *double-* or *two-flow condensers* because the circulating water traverses the length of the condenser twice. By introducing more partitions into the water boxes the condenser may be converted into a *three-flow* or even a *four-flow* condenser. The velocity of the water is made greater the greater the number of flows. The rate of transmission of heat through the tubes to the circulating water increases as the velocity of the water increases but the power required to circulate the water is also increased.

Condenser tubes are made of brass and are most commonly $\frac{3}{4}$ inch in external diameter and about $\frac{1}{20}$ inch thick. Tubes having an external diameter of $\frac{5}{8}$ inch are also frequently used.

Fig. 566 shows in detail the method of connecting the tube plate T

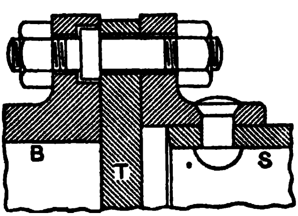


FIG. 566.

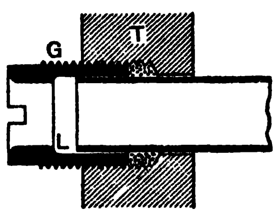


FIG. 567.

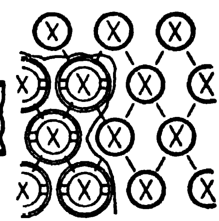


FIG. 568.

to the shell S and also the connection of the water box B to both. It will be seen that the water box may be removed without disturbing the joint between the tube plate and the shell. Condenser tube plates are made of brass, and are generally 1 inch or $1\frac{1}{8}$ inches thick.

Fig. 567 shows how the tubes are connected to the tube plates. The hole for each tube is enlarged for the greater part of its depth and screwed. A ring of cotton tape is slipped over the end the tube after the tube is in position and compressed by the screwed gland G. This design of connection allows the tube to expand or contract freely, while it is at the same time steam and water tight. An internal lip L on the gland prevents the tube creeping out.

The tubes are pitched zigzag on lines at 60° as shown in Fig. 568.

331. Amount of Water for Condensation of Steam.—Consider 1 lb. of steam as it enters the condenser. Let L be its latent heat, x its dryness fraction, t its temperature, and h its sensible heat. Let this steam be condensed in the condenser to water whose temperature is t_1 and sensible heat h_1.

The heat given up by the steam is $xL + h - h_1$.

Let W be the weight, in lb., of the injection water in the case of a jet condenser, or cooling water in the case of a surface condenser, required for the condensation of 1 lb. of steam, and let t_2 and t_3 be the initial and final temperatures of this water, and h_2 and h_3 the corresponding sensible heats. The heat taken up by this water is $W(h_3 - h_2)$.

Assuming that all the heat lost by the steam in condensing is given to the injection or cooling water—

$$W(h_3 - h_2) = xL + h - h_1 \quad \text{and} \quad W = \frac{xL + h - h_1}{h_3 - h_2}$$

The drop or increase in sensible heat may be taken with sufficient accuracy as the temperature difference, then

$$h - h_1 = t - t_1, \text{ and } h_3 - h_2 = t_3 - t_2, \text{ hence } W = \frac{xL + t - t_1}{t_3 - t_2}$$

At the pressures at which steam usually enters a condenser the latent heat L does not exceed 585 C.H.U. (1053 B.Th.U.) and is not less than 565 C.H.U. (1017 B.Th.U.). The average value of L is about 575 C.H.U. (1035 B.Th.U.).

As to the value of x, this is higher for steam turbines than for reciprocating engines, as is clearly shown in Fig. 553, p. 445. In reciprocating engines expansion at constant volume at the end of the stroke in the low-pressure cylinder may reduce the value of x considerably.

For the purpose of preliminary calculations in connection with the design of a condensing plant x is generally taken as one.

The formula given above applies to both jet and surface condensers, but in the case of a jet condenser the final temperature t_3 of the injection water is equal to the temperature t_1 of the cendensed steam, while in a surface condenser t_3 must be less than the temperature of the steam entering the condenser.

The amount of cooling water required to condense 1 lb. of steam in a surface condenser is generally considerably greater than the amount of injection water required for the same purpose in a jet condenser.

332. Vacuum Efficiency.— The minimum absolute pressure at the steam inlet to a condenser is that corresponding to the temperature of the condensed steam, and the corresponding vacuum is the maximum obtainable in a perfect condensing plant, with no air present, at that temperature. The ratio of the actual vacuum obtained at the steam inlet to the condenser to this maximum is called the *vacuum efficiency*.

For example if the temperature of the condensed steam is 35° C., the corresponding absolute pressure of the steam is 0·814 lb. per square inch, and the corresponding vacuum (barometer 30 inches) is $(14{\cdot}74 - 0{\cdot}814) \times 2{\cdot}036 = 28{\cdot}35$ inches. If the actual vacuum at the steam inlet to the condenser is 27·7 inches, the vacuum efficiency is $\frac{27{\cdot}7 \times 100}{28{\cdot}35} = 97{\cdot}7$ per cent.

333. Condenser Efficiency.—There is no standard method of ascertaining the efficiency of a condenser. According to the method

adopted by Messrs. C. A. Parsons & Co., the thermal efficiency of a condenser is the ratio of the difference between the outlet and inlet temperatures of the cooling water to the difference between the temperature corresponding to the vacuum in the steam space and the inlet temperature of the cooling water.

For example: Let the inlet temperature of the cooling water be 28° C. and the outlet temperature 35° C., and let the vacuum be 28·25 inches (barometer 30 inches).

The absolute pressure corresponding to the vacuum is $(30 - 28{\cdot}25) \times 0{\cdot}491 = 0{\cdot}859$ lb. per square inch, and the corresponding temperature is 36° C.

Hence the condenser efficiency is $\dfrac{35 - 28}{36 - 28} = 0{\cdot}875$ or 87·5 per cent.

334. Sources of Air in Condensers--The air found in condensers comes in, (1) with the steam from the boiler into which it enters dissolved in the feed-water, (2) by leakage from the atmosphere at the various joints of parts which are internally under a pressure less than that of the atmosphere, and also through porous castings, and (3), in the case of jet condensers, with the injection water in which it is dissolved.

Feed-water from condensed steam through a circuit which is mainly shut off from the atmosphere, and which is used over and over again, becomes to a large extent purged of air. Hence, in addition to the great saving of heat due to feeding the boiler with condensed steam from the engine, there is the additional advantage of having to deal with less air in the condenser.

Air arriving in the condenser through leakage from the atmosphere may be reduced to a very small quantity by good design, accurate workmanship, and careful attention, during the life of the plant, to the various joints where air leakage may occur.

The amount of air to be found in condensers cannot be stated with certainty, it varies in different cases and at different times, but the following approximate figures may be given.

In the surface condensers of well designed and properly maintained steam turbine plants the amount of air passing into the condenser should not exceed 5 lb. per 10,000 lb. of steam. With reciprocating engines having surface condensers the air leakage is probably 15 lb. per 10,000 lb. of steam. The reason for the greater air leakage in the case of reciprocating engines is that the low pressure glands are not sealed with steam or water, a precaution which is generally taken with turbines.

The air arriving in a jet condenser in the injection water is not usually under control; it is greater in amount the colder the water, and may be taken at about ½ lb. of air to 10,000 lb. of water. If the amount of injection water is, say, 30 lb. per lb. of steam, this means 15 lb. of air per 10,000 lb. of steam, and to this must be added the air which comes in with the steam. It would appear therefore that while the air pump of a steam turbine, having a surface condenser, may have to deal with only about 5 lb. of air per 10,000 lb. of steam, the air pump of a reciprocating engine having a jet condenser may have to deal with about 30 lb. of air per 10,000 lb. of steam.

335. Mixtures of Air and Water Vapour—Dalton's Laws.—Dalton (b. 1766, d. 1844) demonstrated experimentally the following laws which were afterwards verified by Gay-Lussac:—

I. "The pressure, and, consequently, the quantity, of vapour which saturates a given space are the same for the same temperature, whether there is or is not any other gaseous substance in the space."

II. "The pressure of the mixture of a gas and a vapour is equal to the sum of the pressures which each would exert if it occupied the same space alone."

The second law is a consequence of the first, and is known as the *law of partial pressures.*

The application of the above laws to condenser and air pump problems is illustrated by the following examples:—

EXAMPLE I.—The vacuum in a condenser is found to be 28 inches of mercury (barometer 30 inches) and the temperature is 30° C. To find the weight of air present per lb. of uncondensed steam.

From the steam table 1 lb. of steam at 30° C. will exert a pressure of 0·614 lb per square inch and have a volume of 528 cubic feet. The air present per lb. of steam will, by Dalton's first law, have a volume of 528 cubic feet.

The combined pressure of the air and steam is 2 inches of mercury, or $2 \times 0{\cdot}491 = 0{\cdot}982$ lb. per square inch. By Dalton's second law the pressure of the air is $0{\cdot}982 - 0{\cdot}614 = 0{\cdot}368$ lb. per square inch.

For W lb. of air $144pv = 96WT$.

$$\text{Therefore } W = \frac{144pv}{96(273 + 30)} = \frac{144 \times 0{\cdot}368 \times 528}{96 \times 303} = 0{\cdot}962 \text{ lb.}$$

EXAMPLE II.—A surface condenser deals with 5000 lb. of steam per hour and the air leakage amounts to 2 lb. per hour. The temperature of the air pump suction is 35° C. and the vacuum is 27 inches (barometer 30 inches). The volumetric efficiency of the air pump is 80 per cent. To determine the discharging capacity of the air pump, in cubic feet per minute, to remove the air and condensed steam.

At 35° C. the pressure of the steam is 0·814 lb. per square inch.

The combined pressure of air and steam is $(30 - 27) \times 0{\cdot}491 = 1{\cdot}473$ lb. per square inch.

Partial pressure of the air $= 1{\cdot}473 - 0{\cdot}814 = 0{\cdot}659$ lb. per sq. in.

Weight of air per minute $= \frac{2}{60}$ lb.

Volume of air per minute $= v$ cubic feet.

$144 \times 0{\cdot}659v = \frac{2}{60} \times 96 \times (273 + 35)$, from which $v = 10{\cdot}39$.

Weight of steam condensed per minute $= \frac{5000}{60}$ lb.

Volume of condensate $= \frac{500}{6} \times 0{\cdot}0161 = 1{\cdot}34$ cubic feet.

Theoretical capacity of air pump $= 10{\cdot}39 + 1{\cdot}34 = 11{\cdot}73$ cubic feet per minute.

$$\text{Effective capacity of air pump} = \frac{11{\cdot}73}{0{\cdot}8} = 14{\cdot}66 \text{ cubic feet per min.}$$

336. Independent Drives for Air and Water Pumps.—At one time the air and water pumps of a steam engine invariably formed part of the main engine, and their speeds had therefore a constant ratio to the speed of the engine. In all but small steam power plants it is now usual to drive the various pumps independently by separate steam engines, or turbines, or electric motors.

One important advantage of independent drives for engine pumps is that the pump speeds can be readily adjusted to the work they have to do under varying conditions. For example, the amount of cooling water required by the condenser will depend on its initial temperature, which varies from time to time. Another advantage is that the air and circulating pumps can be started before the main engine, or kept running during a temporary stop, and the main engine will more quickly get under way with full load. A still further advantage is that in large plants having several main engines one or more spare pumping plants may be installed, so that a break down in one will not affect, except temporarily, a main engine.

337. Functions of Air Pumps.—The primary function of an air pump is to remove the air which would otherwise accumulate in the condenser and choke it. Another common but not essential function of the air pump is to remove the water as well as the air from the condenser.

It is becoming increasingly common to have separate pumps for extracting the air and water from a condenser. An air pump which extracts both air and water is called a *wet air pump*, while one which removes the moist air only is called a *dry air pump*. So-called dry air pumps are, however, sometimes charged with water for the purpose of water sealing the valves and cooling the moist air, some of the moisture being condensed, or the water may be necessary to the action of the pump as a mechanism. The water charge in a dry air pump is generally used over and over again.

338. Types of Air Pumps.—Air pumps may be divided into, (1) reciprocating piston or bucket pumps which may be either wet or dry pumps, (2) rotary pumps which are generally dry pumps, (3) water jet pumps which are nearly always wet pumps, and (4) steam jet pumps which are dry pumps.

339. Ordinary Reciprocating Air Pump.—Formerly the most common type of air pump was the vertical single acting bucket pump. Fig. 569 shows Messrs. G. & J. Weir's design for this type of pump, which is a wet air pump.

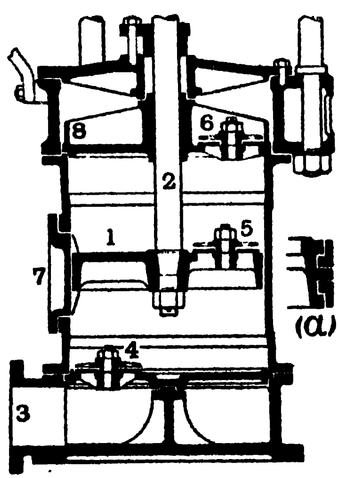


FIG. 569.

The pump shown has an independent drive, the bucket 1 being connected directly to the piston of a steam cylinder by the rod 2, as in the same firm's feed pump illustrated on p. 177. Connection with the condenser is made at 3. Foot, bucket, and head or delivery valves are shown at 4, 5, and 6 respectively. There are of course several valves of each class, but only one of each class is shown. The bucket is provided with metallic packing rings as shown to a larger scale at (*a*). An inspection door 7 gives access to the foot and bucket valves, but

only when the pump is not working. The ledge 8 surrounding the head valves ensures that they are water sealed, so that if there is any leakage through the valves into the pump it will be a leakage of water and not air.

During the up stroke of the bucket, water and air from the condenser enter at 3 and pass upwards through the foot valves, while the charge of water and air above the bucket is discharged through the head valves, the bucket valves being closed. During the down stroke the foot and head valves are closed and the charge below the bucket passes through the bucket valves into the space above the bucket.

Air pump valves are now generally thin metal discs whose seats are gratings surrounding a central stud to the top of which is secured a guard to limit the lift of the disc.

340. Edwards Air Pump.—A design of reciprocating air pump, which has been largely used, and which is highly efficient, is the *Edwards air pump* shown in Fig. 570. This pump has no foot or bucket valves, and since there is no passage through it the bucket becomes a piston. There are head or delivery valves of which one is shown at 1. These valves are placed in the cover 2, which is on top of the pump barrel 3. The piston 4 has a conical bottom 5, and the bottom of the pump casing follows the contour of this part of the piston. Circumferential grooves formed on the piston hold water and form a labyrinth packing. Ports 6 are made in the barrel near its lower end. Connection with the condenser is made at 7.

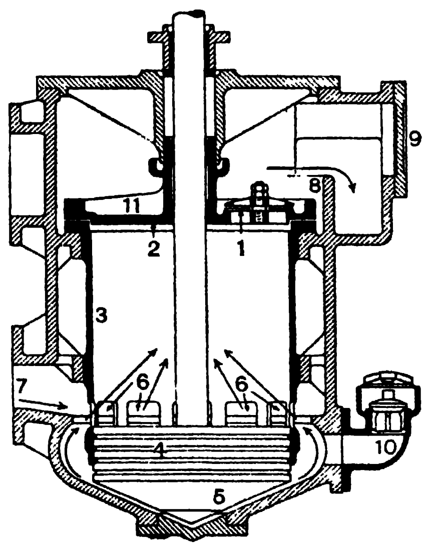


FIG. 570.

During the down stroke of the piston a vacuum is produced in the barrel above the piston, the head valves being closed, and as soon as the piston begins to uncover the ports 6 the air and water vapour from the condenser rush into the space above the piston. The conical part of the piston enters the water which has flowed into the bottom of the pump from the condenser and drives it over the lower part of the barrel and through the ports 6. The pump is now charged, and during the up stroke of the piston the charge is expelled through the head valves and then over the weir 8 on its way to the hot-well. The weir 8 ensures a head of water over the head valves so that if there is any leakage through these valves into the barrel it is a leakage of water and not air.

The head valves are accessible, even when the pump is working, through the door 9.

The water-sealed relief valve 10 opens to the atmosphere when for any reason the pressure below the piston exceeds the pressure of the atmosphere.

The cover 2 is stiffened by having radial ribs 11 cast on it between the head valves.

341. Piston Dry Air Pump with Slide Valve.—A piston pump for withdrawing the moist air without the condensed steam is shown

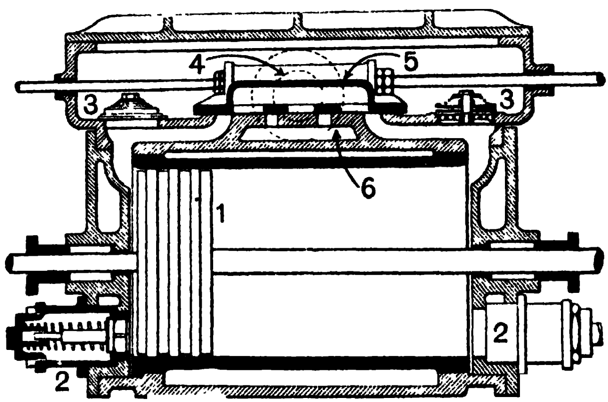


FIG. 571.

in Fig. 571. The piston 1 is shown at one end of its stroke, and it will be seen that the cylinder clearance is small. It is important in a pump of this type that water should be excluded as far as possible. To guard against the presence of water, relief valves 2 are provided in the cylinder covers. The air is discharged through the ordinary lift valves 3 into the discharge pipe 4. A special feature of this pump is the mechanically operated slide valve 5. The functions of the slide valve are to open communication between the two ends of the cylinder for a short period at the end of each stroke of the piston, and to act as a suction valve.

As shown in Fig. 571 the suction passage 6 is closed, and the slide valve is just beginning to uncover the left-hand suction port and make connection between the two ends of the cylinder through the interior of the slide valve. The effect of this is best seen from the indicator diagram in Fig. 572, for which the author is indebted to a paper by Mr. W. A. Dexter. But for the action of the slide valve in equalizing the pressure on the two sides of the piston at the end of the stroke, the compressed air in the clearance space would expand during the next stroke as shown by the dotted curve CE, and no air would be taken in until the piston reached the position E. But the action of the slide valve allows the pressure to drop at once to D and the air then expands to A, which is the position

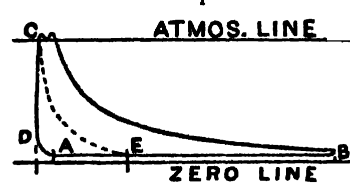


FIG. 572.

of the piston when suction begins. Without the slide valve the effective suction stroke would be EB, and with the slide valve it becomes AB. The volumetric efficiency of the pump is therefore considerably increased.

342. Combined Wet and Dry Air Pumps.—The advent of the steam turbine, which can utilize a high vacuum more efficiently than a reciprocating engine (Art. 316, p. 444), has stimulated endeavour to reduce the pressure in the condenser. One means of improving the vacuum is a supplementary air pump whose duty is to extract the greater part of the air and water vapour which would otherwise remain in the condenser if one ordinary air pump only were used. Such a supplementary air pump is called a *dry air pump* because it has

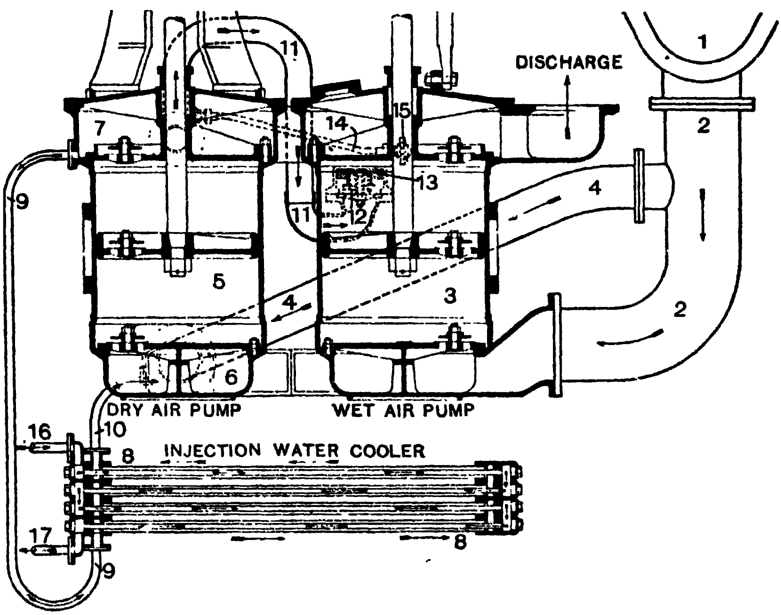


FIG. 573.—Weir dual air pumps.

to deal with a small quantity of water compared with that passing through an ordinary air pump, which is called a *wet air pump*.

The combination of wet and dry air pumps designed and made by Messrs. G. & J. Weir, and known as the *Weir dual air pumps*, is illustrated by Figs. 573 and 574. The former Fig. is partly diagrammatic but the latter shows the actual arrangement in outside elevation. The bottom part of the condenser is shown at 1, and from this the pipe 2 leads the condensed steam to the bottom of the ordinary or wet air pump 3. A branch pipe 4 connects the upper part of 2 to the bottom of the dry air pump 5. The position of the branch pipe 4 in relation to the pipe 2 is such that only the air and water vapour from the condenser can enter the pipe 4. The lower part 6 and the upper part 7 of the dry air pump are charged with

injection water which is cooled by circulating through the surface cooler 8. This cooler is double-tubed and two-flow. The circulating water for cooling the injection water enters the cooler at 16 and flows through the inner tubes leaving the cooler at 17, while the injection water, which is led from the top of the dry air pump to the bottom of the cooler by the pipe 9, flows through the annular spaces between the inner and the outer tubes as shown. The injection water is then led from the top of the cooler to the bottom of the dry air pump by the pipe 10.

The water vapour passing through the pipe 4 from the condenser is condensed by the injection water in 6, and the air coming with the water vapour is cooled. The condensed water vapour and the cooled air and a portion of the injection water are then lifted in the ordinary way by the dry air pump bucket into the space 7 from which the air, and the excess water not required to fill the injection water space, pass through the pipe 11 and through the spring loaded valve 12, and then

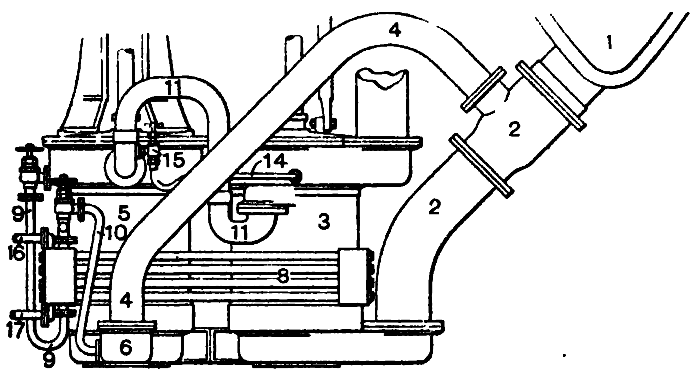


FIG. 574.—Weir dual air pumps.

through the port 13 into the wet air pump which delivers its own charge and that from the dry air pump into the hot-well.

The spring-loaded valve 12 is adjusted to maintain a vacuum of about 20 inches in the dry air pump hot-well when the condenser is working with a vacuum of 28 inches, and this 8 inches difference of pressure is sufficient to cause the injection water to overcome the cooler friction and pass into the suction, and at the same time prevent any direct air connection between the dry suction and discharge.

The charge of injection water for the dry air pump is obtained through the pipe 14, from the hot-well of the wet air pump, by opening the valve 15 for about a minute when the pumps are started.

The pressure range in the dry air pump is less than half that in the wet pump, since the former discharges into the latter during the down stroke of the latter because the buckets move in opposite directions.

Since the circulating water for the cooler does not enter the hot-well from which the boiler feed water is taken, the temperature of the latter is considerably higher than it would be if the same vacuum were produced by means of a single wet air pump. When in efficient

working condition the dry air pump should be at least 15° F. (8·3° C.) below the wet pump temperature.

343. Parsons' Vacuum Augmenter.—A simple substitute for the dry air pump is a steam jet issuing into a convergent combining cone, and this is adopted in *Parsons' vacuum augmenter.* This successful invention is illustrated by Fig. 575. The steam condensed in the main condenser passes direct to the air pump and the greater part of the water vapour and air remaining is extracted by the suction of the steam jet issuing into the convergent combining cone shown. The steam used in the jet is discharged into the augmenter condenser, where it is condensed together with any water vapour accompanying it from the main condenser. The condensate from the augmenter condenser

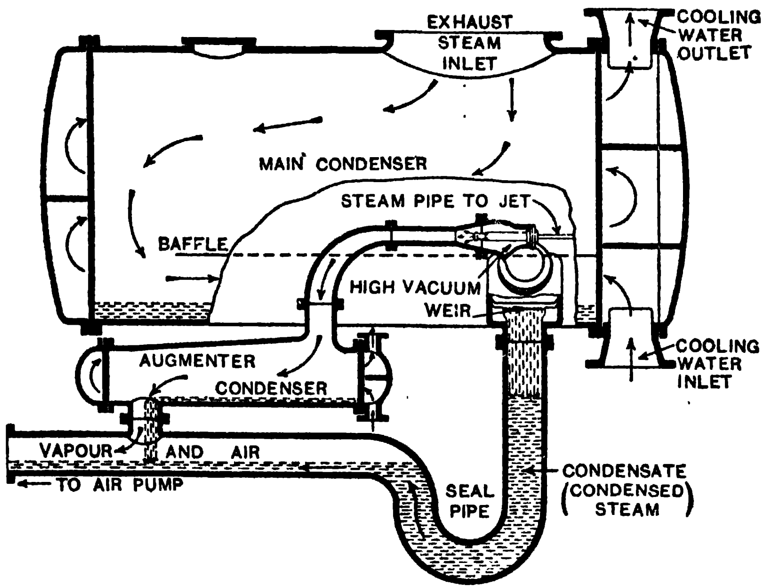


FIG. 575.—Parsons' vacuum augmenter.

together with the air extracted by the steam jet, joins the main condensate and the whole passes to the air pump.

The main and augmenter condensers are surface condensers, but for the sake of clearness the tubes in both have been omitted in the illustration.

The energy of the steam jet is used to force the air from the main condenser into the augmenter condenser where the absolute pressure is higher than in the main condenser. This difference of pressure in the two condensers accounts for the difference of level of the water in the horizontal and vertical parts of the main discharge pipe.

The double bend in the main discharge pipe causes the formation of a water seal which prevents the return of the air extracted by the steam jet.

Messrs. C. A. Parsons & Co. state that the steam used by the vacuum augmenter jet is now only 0·6 per cent. of the steam used by

the turbine at its nominal full load, and the advantages of increased vacuum much more than compensate for the amount of steam which the jet takes.

It should be noted that the load on the air pump is that due to the vacuum in the augmenter condenser and not to the higher vacuum in the main condenser.

Referring to Fig. 575 it will be seen that a weir is provided at the condensate outlet of the main condenser. This weir is a little above the bottom of the condenser and causes enough water to be held back to submerge or drown a few of the bottom rows of tubes through which the coolest circulating water passes. This water which is held back, and which is constantly being renewed, is reduced in temperature and, in consequence, the vapour pressure is also reduced, with the result that the absolute pressure in the condenser is lowered. As a partial set-off against this, the temperature of the boiler feed water is also lowered.

Messrs. C. A. Parsons & Co. inform the author that in the most recent practice ejector air pumps are used, the ejector being simply an amplification of the augmenter principle, in which a compound system of steam jets replaces both the reciprocating air pumps and the augmenter, and no augmenter condenser is employed, but the steam from the ejector is condensed in a feed heater, through which the main boiler feed is circulated. Even when reciprocating air pumps in conjunction with an augmenter jet are installed, the augmenter condenser is not now fitted, as the plant is so arranged that the heat energy in the augmenter steam is recovered in the feed heater, instead of being thrown away in the circulating water, and with this arrangement, there would be no object in "drowning" any of the condenser tubes.

344. Air Pump Capacity in Relation to Vacuum.—Knowing, approximately, the weight of air which an air pump will have to remove from a condenser for a given steam consumption, the capacity of the pump may be computed by the method illustrated in Example II. of Art. 335. Considering this question further, suppose that the condenser temperature is 20° C. (68° F.). The corresponding absolute steam pressure is 0·339 lb. per square inch. If no air were present the vacuum would be $30 - 0{\cdot}339 \times 2{\cdot}036 = 29{\cdot}31$ inches with a barometer of 30 inches.

For a vacuum of 27 inches the total pressure of air and steam is $(30 - 27) \times 0{\cdot}491 = 1{\cdot}473$ lb. per square inch, and the air pressure is $1{\cdot}473 - 0{\cdot}339 = 1{\cdot}134$ lb. per square inch.

Similarly the air pressures for vacua of 28 inches and 29 inches are 0·643 and 0·152 lb. per square inch respectively.

The temperature being the same in each case, the relative volumes of the air to be dealt with by the air pump in the three cases will be inversely as the pressures. Denoting the volume of the air for the 27 inch vacuum by 1, the volumes of the air for the 28 inch and 29 inch vacua will be 1·76 and 7·46 respectively. This shows that the required capacity of the air pump increases rapidly as the theoretical maximum vacuum for a given temperature is approached.

Since the speed of reciprocating air pumps is very limited they become very bulky for high vacua and large powers. This has led engineers to design and experiment with rotary air pumps and steam

air ejectors, examples of which are illustrated and described in the next two Arts.

345. Leblanc Rotary Dry Air Pump.—The air pump shown in Fig. 576, invented by Professor Maurice Leblanc, is selected as an example of a rotary air pump. This is a dry air pump, although it is charged with water for the purpose of its operation.

A cross-section of the pump is shown at (*a*) and a longitudinal section of the upper half at (*b*). A wheel or impeller 1, keyed to a shaft, carries a ring of vanes 2 which is overhung. The pump chambes 3, which is inside the ring of vanes is charged with water which escaper through the nozzle 4 on to the vanes. As the wheel revolves the water from the nozzle is caught up between the vanes and thrown out by centrifugal force in thin sheets which fly into the collecting cone 5 with a velocity of about 130 feet per second. These sheets of water become pistons in the collecting cone, and the air from the condenser coming in at 6 is drawn downwards and entrained between the successive sheets of water. After passing through the collecting cone, the air and water enter the diffuser 7, only a portion of which is shown, where the velocity is gradually reduced and the kinetic energy converted into pressure energy, so that by the time the air and water reach the outer end of the diffuser, which is a divergent pipe, the pressure is sufficient to overcome the pressure of the atmosphere, and, if necessary, a head of water in addition.

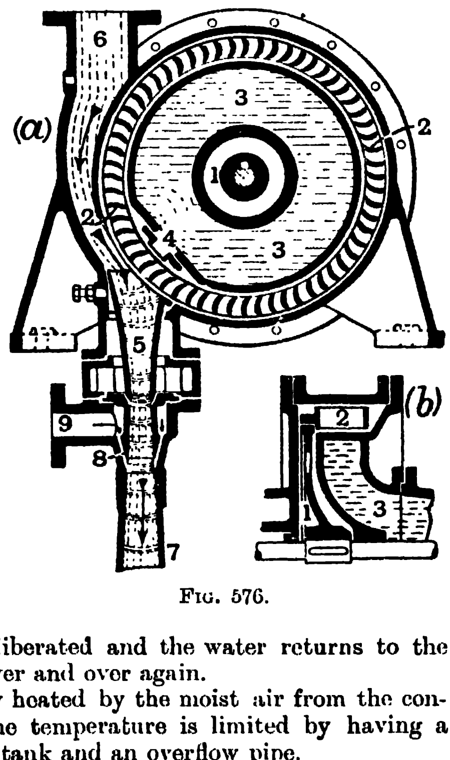


FIG. 576.

The discharge takes place into a tank where the air is liberated and the water returns to the pump chamber 3 and is used over and over again.

The water becomes slightly heated by the moist air from the condenser, and by friction, but the temperature is limited by having a small cold water supply to the tank and an overflow pipe.

If the water level in the tank is above the level of the pump the latter will always be charged with water, but this requires slightly more power to drive the impeller on account of the head of water on the discharge. Without this head of water the pump may be recharged at starting by means of high-pressure steam supplied to the nozzle 8 through the branch 9.

The shaft carrying the impeller is, generally, most conveniently driven by an electric motor.

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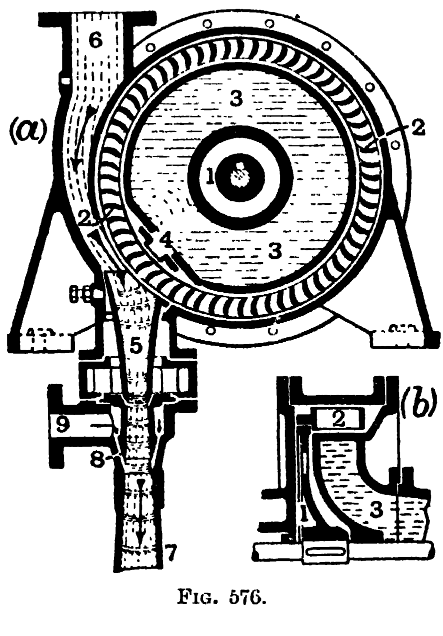

FIG. 576.

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The water becomes slightly heated by the moist air from the condenser, and by friction, but the temperature is limited by having a small cold water supply to the tank and an overflow pipe.

If the water level in the tank is above the level of the pump the latter will always be charged with water, but this requires slightly more power to drive the impeller on account of the head of water on the discharge. Without this head of water the pump may be recharged at starting by means of high-pressure steam supplied to the nozzle 8 through the branch 9.

The shaft carrying the impeller is, generally, most conveniently driven by an electric motor.

346. Steam Air Ejectors.—Very good results are now obtained by using jets of steam to withdraw the air from condensers. Although this method is an old one it is only in recent years that it has been perfected. One of the successful applications of this method is found in *Leblanc's steam air ejector*, an example of which is shown in Fig. 577. The moist air from the condenser is drawn in at 1 and then compressed and discharged at not less than atmospheric pressure. The compression takes place in two stages. In the first stage, in the example shown, there is one steam nozzle 2 and in the second stage there is a group of steam nozzles 3. All the steam nozzles are of the de Laval type.

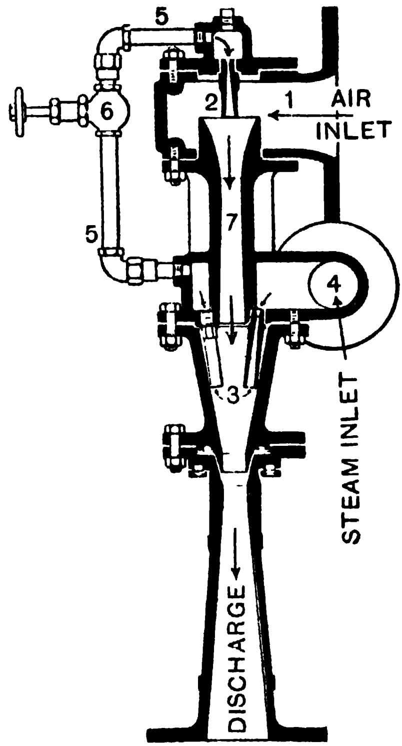


FIG. 577.

Steam, generally at a gauge pressure of not less than 100 lb. per square inch, enters at 4 through a stop valve, not shown in Fig. 577, and feeds directly the second stage nozzles 3. The steam reaches the first stage nozzle 1 through the pipe 5 in which there is a controlling valve 6.

The steam expanding in the nozzles issues from them with a velocity of from 3000 to 4000 feet per second and at a small absolute pressure depending on the vacuum in the condenser. The friction between the first stage steam jet and the surrounding moist air causes the latter to be entrained, and the mixture of steam and air rushes forward into the diffuser 7 where the kinetic energy of the mixture is converted into pressure energy and, before reaching the second stage, both air and steam are compressed to about seven times the pressure in the space 1. The further compression required takes place in the second stage, aided by the action of the steam jets from the nozzles 3.

Discharge takes place into the boiler feed tank where the steam is condensed and its heat utilized.

The steam air ejector is characterized by extreme simplicity and it occupies a very small space compared with other forms of air pumps.

Exercises XXI

1. The vacuum in a condenser is 28·1 inches when the barometer is 30·3 inches. In another case the vacuum is 27·8 inches when the barometer is 29·6 inches. Correct these vacuum readings to the standard barometer of 30 inches.

2. The vacuum at the steam inlet to a condenser was found to be 28·2 inches (barometer 30 inches) and the hot-well temperature was 32° C. The steam pressure corresponding to 32° C. is 0·688 lb. per square inch. Find the vacuum efficiency in the above case.

3. The following data were obtained from a test of a surface condenser: Inlet temperature of circulating water, 12° C. Outlet temperature, 33° C. Vacuum, 27·83 inches (barometer, 30 inches). Compute the efficiency of the condenser.

4. Exhaust steam having a dryness fraction of 0·9 enters a surface condenser at an absolute pressure of 2 lb. per sq. in. and is condensed to water at 35° C. (95° F.). The circulating water enters at 20° C. (68° F.) and leaves at 30° C. (86° F.). Compute the weight of circulating water required per lb. of steam.

5. Taking the data of the preceding exercise, except that the final temperature of the cooling water is to be taken so as to apply to a jet condenser, compute the weight of injection water required per lb. of steam.

6. The following particulars relate to a test of the surface condenser of a steam turbine.—Absolute pressure of the steam entering the condenser, 1 lb. per sq. in. Temperature of condensed steam 30·5° C. Temperature of circulating water, at inlet, 10·2° C., at outlet, 25·7° C. Weight of cooling water per lb. of steam, 30·1 lb. Assuming that all the heat lost by the steam is taken up by the circulating water, determine the dryness fraction of the steam as it enters the condenser.

7. The vacuum in a condenser is 27·3 inches (barometer 30 inches), and the temperature of the steam is 30° C. What is the partial pressure of the air present? If the net volume of the steam space is 400 cubic feet, what is the weight of air in the condenser?

8. The air leakage into a surface condenser and its connections amounts to 4 lb. per 10,000 lb. of steam used. The steam consumption is 28,800 lb per hour. The vacuum in the air pump suction is 28·1 inches (barometer 30 inches), and the temperature is 35° C. Compute the capacity of the air pump, which removes both the air and the condensed steam, in cubic feet per minute, taking the volumetric efficiency as 80 per cent.

9. A jet condenser has to condense 8,400 lb. of steam per hour. The volume of injection water used is 10,200 cubic feet per hour and its initial temperature is 25° C. The volume of air, at atmospheric pressure, dissolved in the injection water is 5 per cent. of the volume of the water, and the air which comes in with the steam and leaks into the condenser amounts to 1 lb. per 2000 lb. of steam. The vacuum in the air pump suction is 26·5 inches (barometer 30 inches), and the temperature of the condensate is 35° C.

Determine the suction capacity of the air pump, in cubic feet per minute, to remove the air and water from the condenser. Take the volumetric efficiency of the air pump as 80 per cent. and the weight of 1 cubic foot of air at 0° C. and atmospheric pressure, 14·7 lb. per sq. in., as 0·0807 lb.

10. The temperature of the steam entering a surface condenser is 50° C., and the temperature of the air pump suction is 45° C. The barometer reading is 29·8 inches of mercury. Find (1) the condenser vacuum, (2) the vapour pressure and the air pressure near to the air pump suction. If the effective capacity of the air pump, on the suction stroke, is 300 cubic feet per minute, find the weight of air entering the condenser per minute, and the weight of steam carried over per minute in the air discharged from the air pump. Assume that for air $PV = 96T$. [U.L.]

11. The air pump for the removal of the air and condensed steam from a surface condenser is single acting and has a diameter of 16 inches and a stroke of 24 inches. The pump makes 60 double strokes per minute. The weight of the steam condensed per minute is 145 lb. The absolute pressure in the air pump suction is 0·85 lb. per sq. in. and the temperature is 32° C. The vapour pressure corresponding to 32° C. is 0·688 lb. per sq. in. Taking the volumetric efficiency of the air pump as 0·75, compute the weight of air passing through the air pump per 10,000 lb. of steam condensed.

12. Assuming a condenser temperature of 25° C. (77° F.), find the relative capacities of a dry air pump necessary for vacua of 26 inches, 27 inches, and 28 inches (barometer 30 inches).

CHAPTER XXII

INTERNAL COMBUSTION ENGINES

347. Distinctive Features of Internal Combustion Engines.—The most prominent feature of an internal combustion engine which distinguishes it from other heat engines is that the heat, part of which is to be converted into work in the engine, is generated within the cylinder of the engine. The heat necessary for a steam engine is generated in the furnace of the boiler, and this heat, or part of it, is carried to the engine in the steam which it uses. In the internal combustion engine the combustion of the fuel takes place in the cylinder and the products of combustion and usually a considerable amount of excess air not required for the combustion of the fuel take the place of the steam in a steam engine.

As a consequence of the combustion taking place in the cylinder of an internal combustion engine, very high temperatures are produced in that cylinder, and to prevent injury to the metal of the cylinder and valves, and to permit of the lubrication of the cylinder, it is necessary to abstract some of the heat from the cylinder, and in the case of large cylinders, from the piston and valves also. This cooling of the cylinder may be effected by the surrounding air in the case of a motor bicycle or an aeroplane engine, otherwise cooling by water made to circulate through jackets surrounding the cylinder barrel and the cylinder head which contains the valves, is the method adopted. Water cooling is the only method used for large valves. Large pistons are generally water cooled, but oil is sometimes used for this purpose, any leakage at the joints is then a leakage of oil and lubrication is more perfect.

Whereas reciprocating steam engines are nearly always double acting, internal combustion engines are most commonly single acting. This is due to the difficulties introduced by the use of stuffing-boxes for the piston rods in internal combustion engines, and to the complication involved in locating the valves in the front end of the cylinder. All double acting internal combustion engines are of large size.

As would be expected, the thermal efficiency of internal combustion engines is much higher than that of steam engines; in fact, as an instrument for converting heat into work the most efficient internal combustion engine is more than twice as efficient as the most efficient steam plant.

348. Cycles of Operation.—The operations which are performed in an internal combustion engine are: (1) Taking in a charge of air for the combustion of the fuel, or (1*a*) taking in a charge of air mixed with

gas or fuel vapour. (2) Compression of the charge. (2*a*) Injection of the charge of fuel when the charge before compression consists of air only. (3) Ignition of the compressed charge of air and fuel. (4) Combustion of the fuel at approximately constant volume, or at approximately constant pressure, or at varying volume and pressure. (5) Expansion of the high pressure products of combustion and heated excess air. (6) Exhausting the spent charge.

These various operations are generally performed either during four strokes, or during two strokes of the piston.

The operations and the number of strokes required for the performance of a complete series of them give rise to various *cycles of operation* the most common of which are explained in succeeding Articles.

349. Otto Four-Stroke Cycle.—The most common cycle of operations in all classes of internal combustion engines is the *four-stroke*

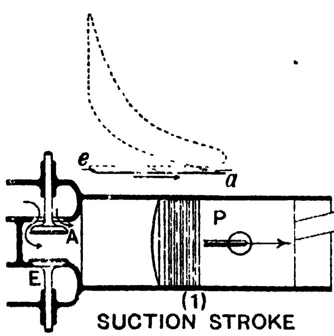


FIG. 578.

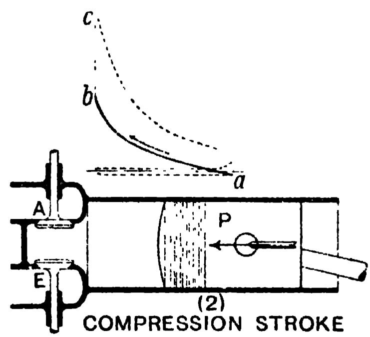


FIG. 579.

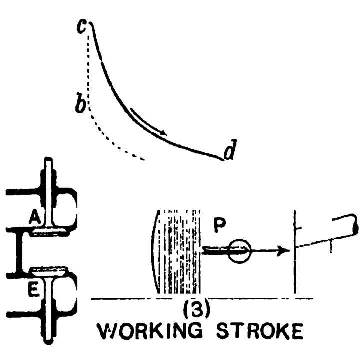


FIG. 580

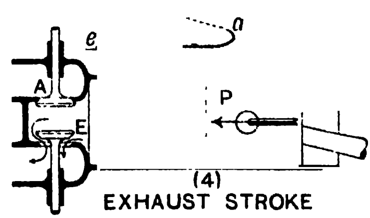


FIG. 581.

cycle. The following description refers to the four-stroke cycle as applied to a gas or petrol engine or any internal combustion engine in which the air for combustion and the fuel in the form of gas or vapour are taken into the cylinder together. The working cylinder is shown diagrammatically in Figs. 578 to 581. A is the admission valve, E the exhaust valve, and P the piston. In the case of gas engines there

is also a gas valve opening into the passage leading to the admission valve A which may also be called the air valve.

Besides the ignition of the charge there are four distinct operations each of which takes up one stroke, more or less.

(1) *Suction stroke.* (Fig. 578).—The exhaust valve is closed, the admission valve is open, the piston is moving outwards, and the charge is taken in. This operation is represented by the line *ea* on the indicator diagram.

(2) *Compression stroke.* (Fig. 579).—Both valves are closed, the piston is moving inwards and the charge is compressed. This operation is represented by the line *ab* on the indicator diagram.

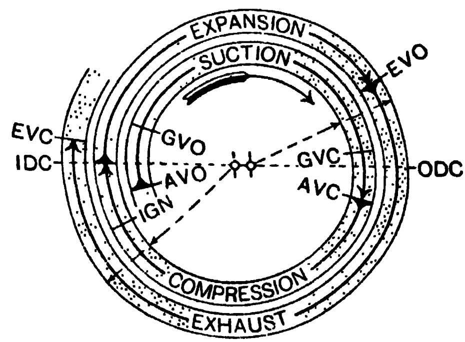
At or very near the end of this stroke the charge is ignited by electric spark, combustion takes place, and the pressure rises as shown by the line *bc* on the indicator diagram.

(3) *Working stroke.* (Fig. 580).—Both valves are closed, and the exploded charge expands driving the piston outwards. This operation is represented by the line *cd* on the indicator diagram.

The exhaust valve opens before the end of this stroke and the exhaust begins.

(4) *Exhaust stroke.* (Fig. 581).—The admission valve is closed, the exhaust valve is open, and the piston is moving inwards and the contents of the cylinder are driven out through the exhaust passage. This operation is represented by the line *de* on the indicator diagram.

The cycle of operations is also shown by the diagram in Fig. 582, called the *valve setting diagram.* This diagram shows the angular positions of the engine crank when the various operations begin and end. Usually this diagram is drawn round one centre, but by using two centres, as shown, the sequence of the different operations is better indicated.



IDC, inner dead centre. ODC, outer dead centre.
AVO, air valve opens. GVO, gas valve opens.
AVC, air valve closes. GVC, gas valve closes.
EVO, exhaust valve opens. IGN, ignition.
EVC, exhaust valve closes.

FIG. 582.

350. Diesel Four-Stroke Cycle. -An engine working on the Diesel four-stroke cycle has, for normal working, three valves, the air admission valve, the fuel valve, and the exhaust valve.

As in the Otto four-stroke cycle, there are four distinct operations in the Diesel four-stroke cycle. The operations are as follows.—

(1) *Suction stroke.*—The exhaust and fuel valves are closed, the air valve is open, and the piston is moving outwards. A charge of air is taken in at nearly atmospheric pressure. This is represented by the line *ea* on the theoretical indicator diagram, Fig. 583.

(2) *Compression stroke.*—All the valves are closed, and the piston

is moving inwards. The air taken in during the suction stroke is compressed to a high pressure (about 500 lb. per square inch). This is represented by the line *ab* in Fig. 583.

(3) *Working stroke.*—The air and exhaust valves are closed, the fuel valve is opened just before the beginning of the stroke and remains open during a small part of the stroke. The heat produced by the high compression raises the temperature of the air sufficiently to ignite the fuel as soon as it is injected into the cylinder, and combustion goes on at least as long as the fuel valve is open. Theoretically the combustion takes place at constant pressure and is represented by the line *bc* in Fig. 583. The products of combustion and the heated excess air now expand as shown by the line *cd* in Fig. 583. Towards

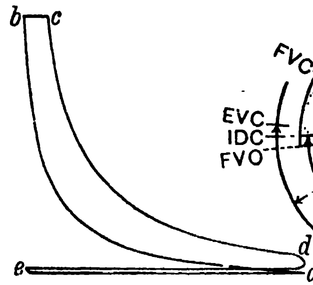


FIG. 583.

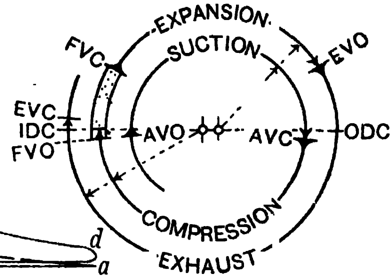


FIG. 584.

IDC, inner dead centre.
ODC, outer dead centre.
AVO, air valve opens.
AVC, air valve closes.
FVO, fuel valve opens.
FVC, fuel valve closes.
EVO, exhaust valve opens.
EVC, exhaust valve closes.

the end of the stroke the exhaust valve opens and the contents of the cylinder begin to leave it.

(4) *Exhaust stroke.*—The air and fuel valves are closed, the exhaust valve is open, the piston moves inwards, and the remainder of the spent charge escapes.

The cycle is further illustrated by the valve setting diagram Fig. 584.

351. Two-Stroke Cycle.—In the *two stroke cycle* the operations of suction, compression, expansion, and exhaust take place in the engine cylinder during two strokes of the piston. Hence, in the two-stroke cycle there is a working stroke for *one* revolution of the crank shaft instead of a working stroke for *two* revolutions in the four-stroke cycle.

The two-stroke cycle engine requires the addition of a pump to force the charge of air and gas, or air only in the case of a two-stroke cycle oil engine, into the cylinder. In the later forms of two-stroke cycle gas engines two pumps are used, one for air and the other for gas.

In practically all two-stroke cycle engines the exhaust takes place through ports in the cylinder, which are uncovered by the piston towards the end of the out-stroke.

The operation of the two-stroke cycle engine will be understood by reference to Fig. 585, which shows the working cylinder diagrammatically. A is the admission valve, E the exhaust ports, and P the piston. The exhaust ports lead into a hollow belt passing round the cylinder.

It will be simplest to describe the cycle beginning at the point where the piston is just about to uncover the exhaust ports during the out-stroke. When the piston is in this position the space behind it contains the expanded charge at a pressure indicated by the point *a* on the indicator diagram. During the completion of its out-stroke the piston uncovers the exhaust ports and the expanded charge escapes. In the case of the two-stroke cycle gas engine the admission valve opens when the piston has reached the end of its out-stroke, the pressure in the cylinder being then indicated by the point *b*, and the fresh charge enters at a gauge pressure of about 5 lb. per square inch. When there are separate pumps for air and gas, air enters first followed by gas and more air. In the case of the two-stroke cycle oil engine the admission valve opens at *a*, as soon as the piston begins to uncover the exhaust ports. In this case, of course, only air is admitted.

During the in-stroke, compression of the charge of air and gas, or of air alone in the case of the oil engine, begins at *c*, as soon as the exhaust ports are covered by the piston. Compression goes on to the end of the in-stroke as shown by the compression curve *cd*. The charge of gas and air is then ignited and the pressure rises from *d* to *e*. During the following out-stroke expansion takes place as shown by *ea*. The compression, ignition and combustion, and expansion parts of the cycle in the two-stroke cycle gas engine are evidently the same as in the four-stroke cycle engine. This is also true for the oil engine.

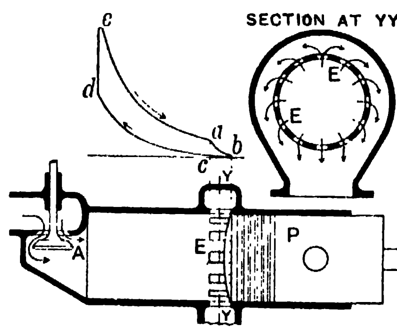


Fig. 585.

There is of course a danger of some of the charge escaping through the exhaust ports while the admission valve is open, but this danger is reduced when there are separate pumps for air and gas, because then it can be arranged that the air precedes the gas and any loss through the exhaust ports is more likely to be a loss of air only, but this escaping air has a scavenging effect, which is advantageous.

In the two-stroke cycle oil engine the air pump for the main air supply is generally called the scavenging pump.

352. Indicated Horse-Power of Internal Combustion Engines.—The maximum pressure in the cylinder of an internal combustion engine being very high a very stiff spring is necessary in the indicator when taking a diagram of the complete cycle, and this stiff spring is not in general sensitive enough to show the negative loop of the diagram, which represents the negative work during the pumping strokes. This loop is therefore ignored in finding the mean height of the diagram or the mean effective pressure on the piston. Even in cases where the negative loop is measurable it is usual to take the mean effective pressure from the positive loop only, the work represented by the negative loop being charged to the friction of the engine.

Let: d = diameter of piston in inches.
l = length of stroke in inches.
p = mean effective pressure, in lb. per square inch.
n = number of working cycles per minute.
N = number of revolutions of crank shaft per minute.

Then, I.H.P. $= \frac{\pi}{4} \times d^2 \times p \times \frac{l}{12} \times n \div 33{,}000$ for each single acting piston. For a double acting piston the work done on the other side of the piston must be added, allowance being made for the area of the piston rod, and if the piston rod goes through both ends of the cylinder allowance for its area must be made on both sides of the piston.

If every cycle is a working cycle, then for a four-stroke cycle $n = \frac{N}{2}$, and for a two-stroke cycle $n = N$.

From the expression for the I.H.P., by transposition, the mean effective pressure is, $p = \frac{4 \times 12 \times 33{,}000}{\pi \times d^2 \times l \times n} \times$ I.H.P.

If for the indicated horse-power I.H.P. in the foregoing expression the brake horse-power be substituted, then p becomes the *brake mean effective pressure*, a quantity which it is convenient to know in cases where it is not easy to get accurate indicator diagrams, as in very high speed engines.

It is evident that the brake mean effective pressure is equal to the indicated mean effective pressure multiplied by the mechanical efficiency of the engine.

353. Ignition.—In internal combustion engines in which the fuel gas or vapour is compressed along with the air for its combustion, the combustion may be started by raising the temperature of the whole charge sufficiently by compression and contact with the hot surface of the combustion chamber, but this method is open to the objection that the point of the cycle at which combustion begins is very uncertain. When, however, the air only is compressed in the cylinder, and the fuel is injected into it at the end of the compression stroke, this method is very satisfactory and is now largely used in oil engines.

In all other cases only a very small part of the mixture of fuel and air is heated and combustion started. This is now done in nearly all new engines by electric spark. It is worthy of note that in the earliest gas engines the electric spark was used for igniting the charge, but this was superseded by flame ignition, which was in turn replaced by the hot-tube. The hot-tube was closed at one end and was of metal or porcelain, and was kept red-hot by an external Bunsen burner. The interior of the hot-tube communicated through a timing valve alternately with the cylinder, when ignition took place, and with the atmosphere, when the compressed products of combustion expanded and escaped.

Electric ignition is performed either by low-tension or by high-tension currents carried to one or more sparking plugs in the combustion space of the cylinder. These currents are generally

obtained by means of a small magneto-electric machine, called a *magneto*, operated by the engine.

In high-tension coil ignition a low-tension current from batteries or accumulator cells or from a magneto is fed to an induction coil, which produces a high-tension current which is carried to the sparking plug. In the Lodge high-tension coil ignition system the electric current from the induction coil, on its way to the sparking plug, is used to charge the inner coatings of two Leyden jars by means of which a very effective spark is obtained, which is not hindered by deposits of oil, soot, etc., on the sparking plug.

354. Governing.—The function of a governor in any heat engine is to regulate the supply of heat energy according to the load on the engine. In all cases the governor itself requires some change of speed before it can act, but the variation in the speed of an engine does not depend only on the sensitiveness of the governor, but also on the way in which the supply of energy to the engine is effected under the action of the governor. The actual governor may be practically the same in all heat engines, although the way in which the energy supply is regulated may be very different in different engines. The subject of governors has been treated fairly fully in Chapter XVII., and in this article only the methods adopted for regulating, under the control of the governor, the power developed in the cylinder of an internal combustion engine will be considered.

The methods of governing internal combustion engines are: (1) Completely cutting off the fuel supply for one or more cycles. This is called the *hit-and-miss* method. (2) Varying the supply of fuel to the cylinder. This is called the *quality* method, because the ratio of air to fuel or the quality of the mixture is altered. (3) Varying the supply of air as well as the supply of fuel, the ratio of air to fuel being kept approximately constant so that the quality of the mixture remains approximately the same. This is called the *quantity* method. (4) A combination of the quality and quanity methods.

Hit-and-miss method.—The mechanism for applying this method to an oil engine is illustrated in Fig. 600, p. 501, and described on p. 502. As applied to a gas engine the mechanism is the same, but it is the gas valve instead of the plunger of the fuel-oil pump which is operated or put out of action. This method is not used on large engines, and on the smaller engines it is not now so common as it has been, notwithstanding that it has several important advantages.

The advantages of the method are as follows: The mixture, once it is adjusted to give the best result, remains practically constant, and in every working cycle the indicator diagram is practically the same, and therefore the same amount of work is obtained from the fuel used The governor, having practically no resistance to overcome, may be of a very light and simple form. Also, the whole mechanism is extremely simple and little liable to get out of order.

The great disadvantage of the method, especially for large engines, is the absence of all turning effort on the crank shaft during the idle cycles, necessitating a very heavy fly-wheel to keep the fluctuation of speed within reasonable limits. Increasing the weight of the fly-wheel

increases the friction of the main bearings, and therefore diminishes the mechanical efficiency of the engine.

Quality method.—As applied to gas engines quality governing is effected by varying the lift of the gas valve, or the time during which it is open, or by varying both lift and time. Another and a simple way is to have a throttle valve in the gas passage leading to the admission valve and operated by the governor.

As applied to oil engines the suction valve of the fuel-oil pump may be kept open by the governor during part of the delivery stroke, the oil being then returned to the supply chamber until the governor closes the suction valve, after which, for the remainder of the stroke, the oil is discharged through the delivery valve. This is the general practice in Diesel engines. Another practice is to vary the stroke of the fuel-oil pump. In another design there is a control valve on the delivery side of the pump which opens under the control of the governor after a part of the stroke of the plunger has been performed, and the oil delivered during the remainder of the stroke returns to the suction side.

In quality governing, there being no restriction as to the amount of air admitted, the same weight of charge is taken into the cylinder during each suction stroke; hence the pressure reached during compression is always the same, but with a diminished fuel supply the maximum pressure reached in explosion engines is lowered. Theoretically the thermal efficiency is, however, unchanged, being dependent on the compression ratio only (see Art. 370, p. 515). At light loads, say less than half-load, with this method of governing the efficiency generally drops because of the difficulty of obtaining prompt combustion with weak mixtures.

In two-stroke cycle engines quality governing is always adopted, as it is important in these engines that the cylinder should be fully charged in order to get rid of the spent charge of the previous cycle.

Quantity method.—This method is not used in oil engines other than petrol engines. It is applied to petrol engines by having a throttle valve in the pipe leading from the carburettor to the cylinder. It is applied to gas engines in various ways. The gas and air supplies may each be throttled by separate valves in the gas and air passages leading to the admission valve or, after passing through a mixing valve, the air and gas together may be throttled by a single valve just before reaching the admission valve. Another method is to regulate the lift of the admission valve. An example of this method of governing is illustrated in Fig. 586, p. 486, and described on p. 485.

In quantity governing the weight of the charge varies, consequently the final compression pressure must vary with the load. This means a diminished thermal efficiency as the load is reduced, but the constant ratio of air to gas in the quantity method favours more efficient combustion at light loads than is the case with the quality method. The result is that at light loads quantity governing is more economical than quality governing.

Combination method.—Where close regulation of speed is required combined with economy of gas, a combination of the quality and quantity methods is generally used. As would be expected, this

requires a more complicated valve gear than is necessary for either method by itself. In combination governing the quality method is used for loads in the neighbourhood of the normal load and the quantity method for the lighter loads.

355. Fuels for Internal Combustion Engines.—The fuel supplied to an internal combustion engine is either gaseous or liquid. Except in the case of *natural gas*, which is available in certain parts of the world, a gaseous fuel is generated either at a central station or in a gas producer specially set up for supplying one or more engines. The central station plant may be one for producing what is called *town gas*, which is used for lighting, heating, and power purposes. The central station plant may, however, be one for supplying gas for heating or power purposes and not for lighting.

The usual combustible constituents of gaseous fuels are, hydrocarbons (chiefly or almost entirely methane, CH_4), hydrogen (H_2), and carbon monoxide (CO). The other constituents are, carbon dioxide (CO_2), and nitrogen (N_2), and a usually negligible quantity of oxygen (O_2).

The calorific value of a gaseous fuel may be determined by burning it in a calorimeter, or it may be calculated from an analysis of the gas. Analyses of gaseous fuels are always by volume, and the various data refer to a cubic foot of the gas at standard pressure and temperature (14·7 lb. per square inch, and 0° C.).

The calorific values of the combustible constituents of gaseous fuels per lb. will be found on p. 105, and their specific densities and specific volumes are given on p. 571.

The following table, computed from the data just referred to, will be found useful in calculating the calorific values of gaseous fuels.

Calorific Values of the Combustible Constituents of Gaseous Fuels per cubic foot at 0° C. and 14·7 lb. per sq. inch.

	Hydrogen (H_2).		Carbon monoxide (CO).		Methane (CH_4).		Ethylene (C_2H_4).	
	C.H.U.	B.Th.U.	C.H.U.	B.Th.U.	C.H.U.	B.Th.U.	C.H.U.	B.Th.U.
Higher	193	348	190	342	598	1076	968	1743
Lower	166	299	190	342	543	978	913	1644

In all internal combustion engine calculations involving the calorific value of the fuel the *lower* calorific value is nearly always taken.

Particulars relating to the principal classes of gaseous fuels are given in the following table. It must be understood, however, that the compositions are subject to variations due to variations in the fuel from which the gas is produced, and to variations in the working of the plant. All the data below the given compositions have been found by calculation.

Particulars relating to Gaseous Fuels.

	Town gas.	Coke-oven gas	Pressure producer gas from bituminous coal.	Suction producer gas from anthracite coal.	Blast-furnace gas.
	Composition, by volume, per cent.				
Hydrocarbons (C_2H_4, etc.)	4	2	—	—	—
Methane (CH_4) . . .	37	30	2	1	—
Hydrogen (H_2)	49	52	27	15	3
Carbon monoxide (CO) .	4	7	12	21	27
Carbon dioxide (CO_2) .	1	2	15	8	9
Nitrogen (N_2)	5	7	44	55	61
Weight of 1 cubic foot . lb.	0·0308	0·0313	0·0647	0·0705	0·0800
Volume of 1 lb. . cu. ft.	32 47	31·95	15·46	14·18	12·50
Calorific value per lb. Higher C.H.U.	11,761	9,993	1348	1063	715
Calorific value per lb. Higher B.Th.U.	21,170	17,988	2427	1914	1287
Calorific value per lb. Lower C.H.U.	10,607	8,982	1211	993	701
Calorific value per lb. Lower B.Th.U.	19,092	16,167	2180	1787	1262
Calorific value per cu. ft. Higher C.H.U.	362	313	87·2	75	57·2
Calorific value per cu. ft. Higher B.Th.U.	652	563	157	135	103
Calorific value per cu. ft. Lower C.H.U.	327	281	78 3	70	56·1
Calorific value per cu. ft. Lower B.Th.U.	588	506	141	126	101
Weight of air for combustion of 1 lb of gas . lb.	14 25	11·88	1·41	1·10	0·73
Vol. of air for combustion of 1 cu. ft. of gas cu. ft.	5·36	4·55	1·12	0·95	0 71

All volumes are taken at 0° C. and 14·7 lb. per sq. in

Density and Calorific Values of Liquid Fuels.

Fuel.	Specific Gravity	Lower calorific values per pound. C.H.U.	Lower calorific values per pound. B.Th.U.	Lower calorific values per gallon. C.H.U.	Lower calorific values per gallon. B.Th.U.	Types of engines in which commonly used (p. 484).
Heavy crude petroleum	0·94	10,600	19,080	99,640	179,352	I.
Light ,, ,,	0·82	10,900	19,620	89,380	160,884	I.
Russian ostatki . .	0·93	10,100	18,180	93,930	169,074	I. & II.
American residuum .	0·89	10,200	18,360	90,780	163,404	I. & II.
Average heavy fuel oil	0·89	10,000	18,000	89,000	160,200	I. & II.
Shale oil	0·87	9,500	17,100	82,650	148,770	I. & II.
Coal tar oil	1 05	9,000	16,200	94,500	170,100	I.
Paraffin oil or kerosene.	0·81	10,200	18,360	82,620	148,716	III. & IV.
Petrol or gasolene .	0·69	10,650	19,170	73,485	132,273	V.
	0·76	10,150	18,270	77,140	138,852	V.
Alcohol (meth. spirit)	0·83	5,500	9,900	45,650	82,170	V.
Benzene	0·88	9,500	17,100	83,600	150,480	V.

The crude petroleum, that is, the petroleum as it comes from the wells, is not used so much for internal combustion engines as the residues obtained after distilling off the petrol or gasolene and the paraffin oil or kerosene from the crude oil. Russian ostatki and American residuum are such residues.

Where fuel oil or spirit is sold by the gallon or where fuel storage space is limited, the calorific value per gallon may be more important than the calorific value per lb.

With reference to the use of alcohol in place of petrol it should be noted that, although the calorific value of the former is much lower than that of the latter, a very much higher compression is permissible with alcohol, without premature ignition, which largely compensates for its lower calorific value.

356. Types of Oil Engines.—In the following approximate classification the engines may work either on the four-stroke cycle or on the two-stroke cycle.

I. *Diesel engines.*—Air only compressed. Temperature due to compression sufficient to ignite the oil fuel, which is injected and sprayed into the cylinder by high-pressure air at the end of the compression stroke.

II. *Semi-Diesel engines.*—Air only compressed, but compression is lower than in Diesel engines and the temperature due to compression is not sufficient to ignite the oil fuel. The fuel is injected (generally, but not always, without the aid of high-pressure air) into a vaporizer on the cylinder head. Part of the vaporizer, called the *hot-bulb*, is unjacketed and its temperature, raised before starting by a blow-lamp, is afterwards maintained by the heat of combustion of the fuel. The mixture of air and fuel vapour is ignited by the hot-bulb. The combustion takes place mainly at constant volume, but partly as in the Diesel engine.

III. *Explosion engines*, in which the oil fuel is vaporized by heat in a vaporizer before entering the cylinder, and this vapour is drawn in and mixed with air during the suction stroke as in a gas engine. The vaporizer is heated before starting by a blow-lamp, and its temperature is afterwards maintained by the heat of combustion of the fuel. The mixture of *air and fuel vapour* is compressed and then ignited by a hot-bulb or its equivalent a hot-tube, the hot-bulb or hot-tube being kept hot by the combustion of the fuel.

IV. *Explosion engines*, in which the vaporizer is heated by the exhaust gases or hot water from the cylinder jacket flowing through a jacket surrounding the vaporizer. Combustion does not take place in the vaporizer. After passing through the vaporizer the fuel enters the cylinder and mixes with the air during the suction stroke. The mixture of *air and fuel vapour* is compressed and then *ignited by electric spark* as in a gas engine.

V. *Explosion engines* using *unheated vaporizers or carburettors.* The mixture of air and fuel vapour is compressed and then ignited by electric spark as in a gas engine.

357. Limiting Compression Pressure—Premature Ignition.—Where the air and fuel gas or fuel vapour are compressed together in the engine cylinder if the compression pressure is too high the resulting

temperature is sufficient to ignite the charge before the end of the compression stroke. This must of course be avoided by limiting the final compression pressure, according to the kind of fuel used, approximately as follows, the pressures being in lb. per square inch: Petrol, 80. Alcohol, 200. Town gas, 130. Producer gas, 170. Blast furnace gas, 220. Heavy fuel oil, 270.

Where the air is compressed separately, as in Diesel and semi-Diesel engines, there is of course no limit to the compression pressure so far as danger from premature ignition is concerned.

The safe compression pressure is generally higher the lower the percentage of hydrogen in the fuel.

With a given fuel the safe compression pressure may be raised by the injection of a small quantity of water into the cylinder or into the vaporizer during the suction stroke.

358. Crossley Gas Engine.—The characteristic features of the Crossley gas engine for medium powers are well shown in Figs. 586 and 587, prepared from drawings kindly supplied by Messrs. Crossley Bros., Manchester. The particular engine illustrated has a single horizontal cylinder 17 inches in diameter and a piston stroke of 26 inches. Running on producer gas made from anthracite, bituminous coal, or wood fuel, the normal load is 85 B H.P. and the maximum load 95 B.H.P. at 190 revolutions per minute. When running on gas made from coke the power is 6 per cent. less.

Referring to the illustrations (Figs. 586 and 587) the first noticeable feature is the admission valve, which consists of two single beat valves 1 and 2, for mixture and gas respectively. These two valves are mounted on the same spindle 3 upon which is clamped the sleeve 4. When the admission valve is open, as shown in Fig. 586, the upper part of the gas valve is in contact with the bottom of the sleeve 4. When the spindle 3 is lifted the gas valve 2 comes to its seat just before the mixture valve 1 closes, and during the completion of the lift the spindle slides through the part 2 against the resistance of an internal spring shown in the enlarged detail section at (A) in Fig. 587.

The valves 1 and 2 are operated as follows. A lever 5 is jointed at one end to the spindle 3 by the pin 6 and at the other end to the rod 7 by the pin 8. The lower end of the rod 7 is jointed to the short swinging piece 9 which carries the roller 10 which is in contact with the admission cam 11 on the half-speed cam shaft. The spindle 3 is pushed upwards by the spring 12 and the lever 5 is pressed against the lower end of the movable fulcrum lever 13 which forms the fulcrum of the lever 5. The distance the spindle 3 is lowered under the action of the cam 11 will evidently depend on the position of the fulcrum 13 of the lever 5 and this is decided by the governor through the mechanism shown and which will be readily understood without further description.

It will be seen that the governing is by the *quantity* method. The ratio of air to gas remains practically constant, but the amount of the charge admitted is regulated as just described by varying the opening of the compound admission valve. The governing mechanism is extremely simple and has proved very effective. The effect of the

governor on the form of the indicator diagram at different loads is shown in Figs. 588 and 589.

The engine is designed to run with a speed variation, from full load to no load, of 1·5 to 2 per cent. above or below the mean speed. By

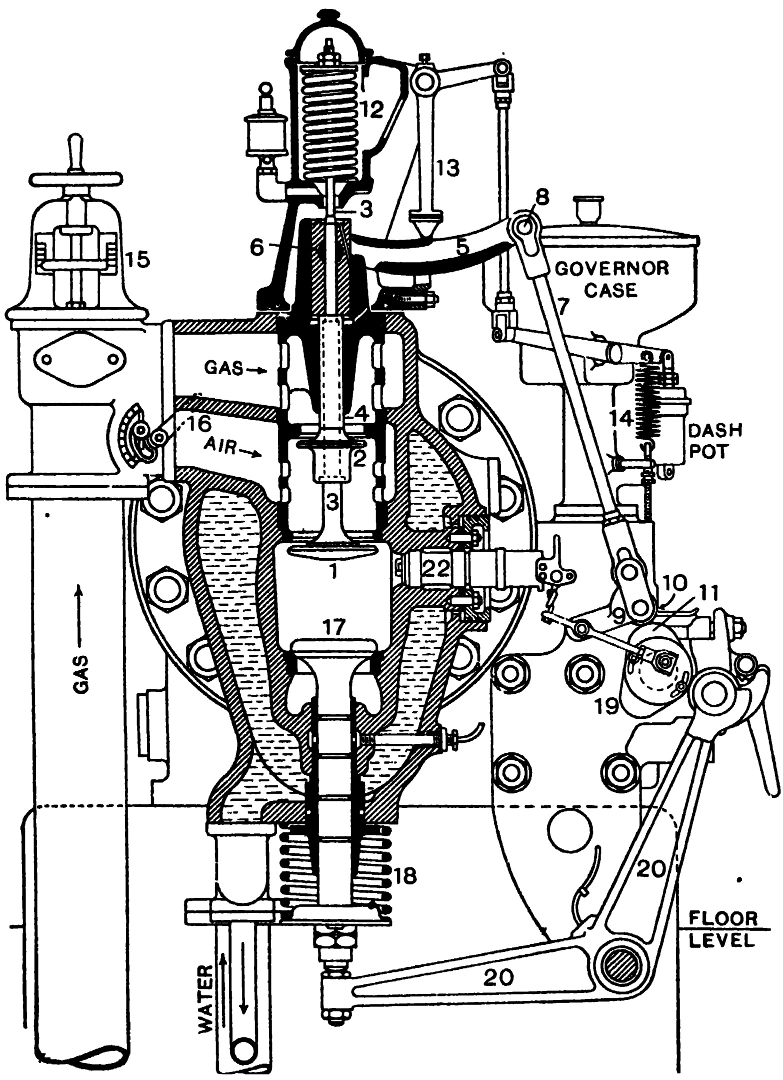


FIG. 586.—Crossley gas engine. Cross section.

means of the adjustable spring 14, which forms a supplementary load on the governor, the speed of the engine may be varied slightly while running.

The gas supply is regulated by hand or shut off altogether by means

of the screwdown valve 15, and the air supply is regulated by a throttle valve in the air passage operated by the hand lever 16.

The exhaust valve 17 is loaded by the spring 18 and is operated

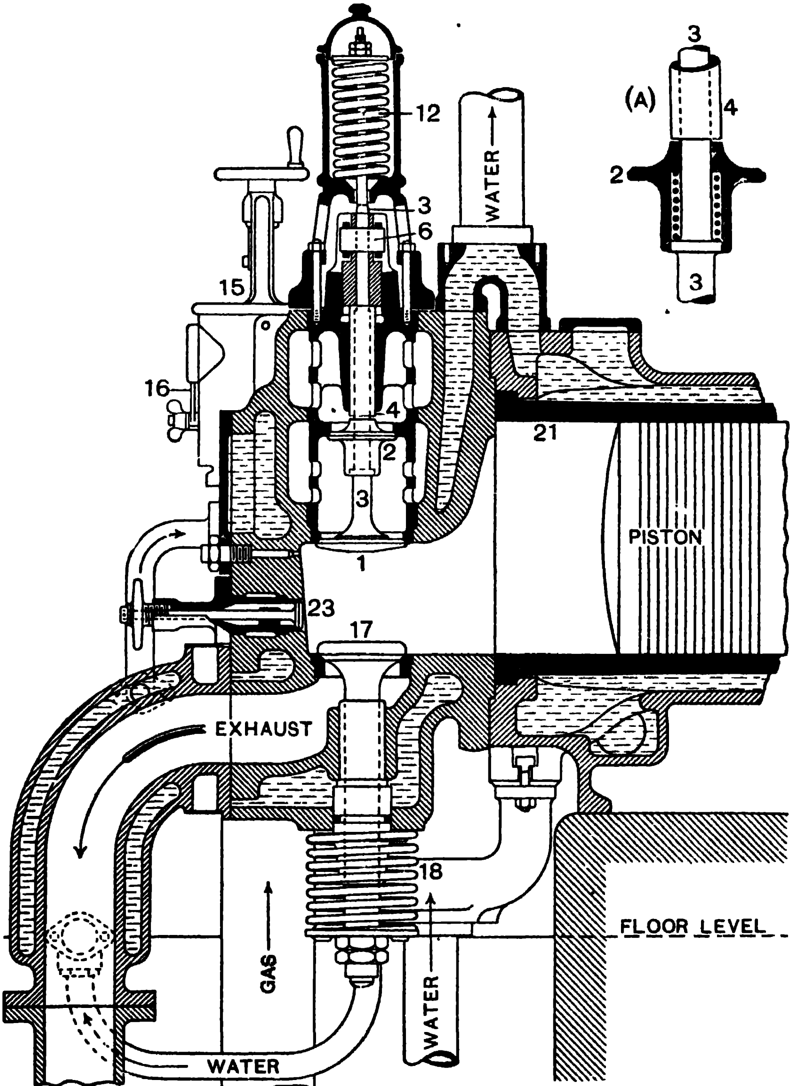


FIG. 587.—Crossley gas engine. Longitudinal section.

by the exhaust cam 19 through the bent lever 20 as shown. It will be seen that the part of the exhaust pipe adjacent to the engine is water jacketed. The cooling water system is clearly shown in the illustrations.

The cylinder liner 21 is clamped between the cylinder head and the main casting which forms the frame of the engine. At the outer end the liner passes from the cylinder jacket by a bored and turned joint in which there are two grooves containing rubber rings to prevent water leakage and allow free longitudinal expansion of the liner.

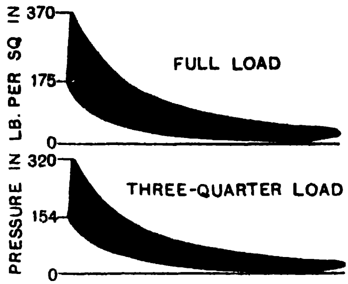


FIG. 588.

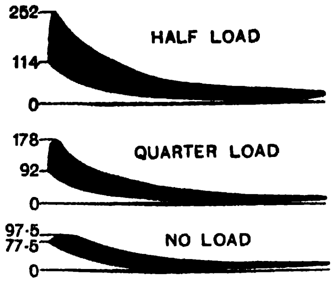


FIG. 589.

Ignition of the charge is by electric spark produced by a low-tension magneto not shown in the illustrations. The sparking plug is shown at 22 in Fig. 586.

The engine is started by compressed air through the starting valve 23 or by a hand pump and petrol cup starter.

359. Körting Two-Stroke Cycle Gas Engine.—A design of gas engine which has been extensively used for large powers is that due to Korting Brothers of Hanover. The chief features of this design are as follows. The engine is double acting and has an inlet valve at each end of the cylinder for admitting air and gas, and there are exhaust ports round the centre of the cylinder which are covered and uncovered by the piston. The air and gas are pumped into the cylinder but they do not mix until they reach the inlet valve, there being separate pumps for air and gas. Moreover, a charge of air alone enters the cylinder through the inlet valve for the purpose of scavenging and this is followed by a charge of air and gas which mix together at the inlet valve. The engine works on the two-stroke cycle, and being double acting, there are two working strokes for each revolution of the crank shaft as in a double-acting steam engine.

The earlier form of this Körting engine is shown diagrammatically in Fig. 590. The inlet valves V are opened by cams and levers and are closed by springs. The exhaust ports E are shown just covered by the piston. The air and gas pumps are double acting and are provided with piston valves P, each piston valve acting as a suction valve and also as a delivery valve. The pistons of the charging pumps are on the same rod and are worked from a crank on the engine crank shaft, the crank which actuates the pumps being set at about 110° in advance of the main crank.

Considering what takes place in one half of the cylinder, the action of the engine is as follows. When the exhaust ports E are uncovered the spent charge escapes and the pressure in the cylinder falls to near that of the atmosphere. Previous to this the passages leading from the

pumps to the inlet valve have been charged, but since the gas delivery valve opens later than the air delivery valve the compressed air behind the inlet valve has forced the gas back some distance in the gas passage and when the inlet valve V opens this compressed air rushes into the cylinder and expels the remaining spent charge through the exhaust ports. A little later the delivery valve of the gas pump opens and the gas enters the cylinder together with the continued supply of air from the air pump until the inlet valve closes. After the inlet valve is closed and the exhaust ports are covered by the piston, compression of the charge begins and is continued to the end of the stroke. Just before the end of the compression stroke the charge is ignited. The next stroke is the working stroke, towards the end of which the exhaust ports are again uncovered.

Fig. 590 shows one arrangement for governing the engine. There

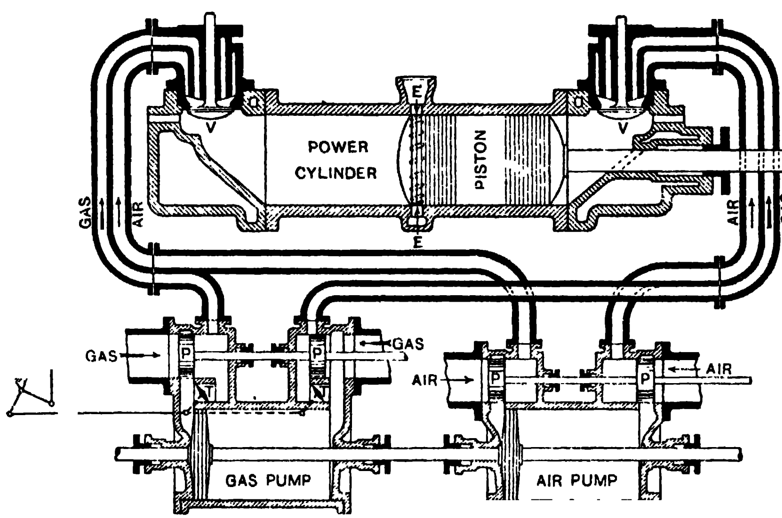


FIG. 590.—Körting double acting two-stroke cycle gas engine.

are by-pass or throttle valves T in passages between the delivery and suction passages of the gas pump. These valves are controlled by the governor. The opening of valves T, more or less, causes the pump to draw gas, during the suction stroke, from the delivery passage leading to the cylinder and its place is taken by air at the inlet valve end of that passage. This action continues for a portion of the delivery stroke of the gas pump. When the inlet valve opens the charge to the cylinder will therefore contain more than the normal quantity of air and the strength of the mixture will depend on the amount the throttle valves T are open.

The Körting engine is now made by a considerable number of firms and modifications and improvements in its design have been made by them.

A longitudinal section of the form of piston used in large gas

engines of the Körting type is shown in Fig. 591. This piston it will be seen is water cooled.

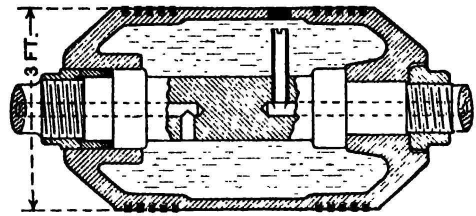


FIG. 591.

360. Oechelhäuser Gas Engine.—Another design of gas engine suitable for large powers is shown diagrammatically in Fig. 592. This is the Oechelhäuser engine, named after its inventor. It is a very simple form of engine and works on the two-stroke cycle.

The cylinder is open at both ends and it contains two pistons FP and BP, which reciprocate, always moving in opposite directions. The pistons also act as valves by covering and uncovering three sets of ports E, A, and G, in the cylinder. The air and gas are forced into the cylinder by a pump which, in Fig. 592, is at the back of the engine and is driven directly from the back piston BP. But two separate pumps may be used, one for air and the other for gas, and they may be placed at the side of the engine and be driven by a separate crank on the crank shaft, or they may be placed below the floor level and be driven from a rocking shaft worked by a lever from the back cross-head.

The pump has suction valves S and delivery valves D. These valves, which are not shown in Fig. 592, are of the automatic type such as are used on air compressors.

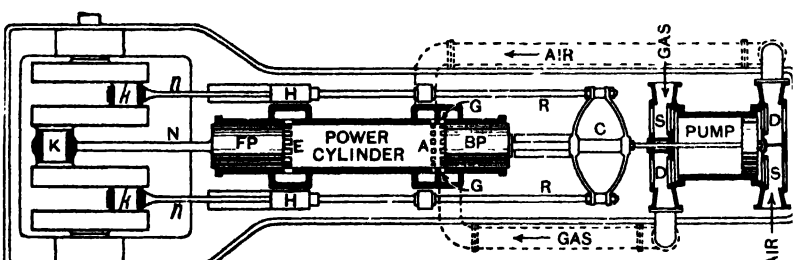


FIG. 592.—Oechelhäuser gas engine.

The front piston FP is coupled to the crank pin K of the central crank by means of the connecting rod N. The back piston BP is coupled to the crank pins *k* of the outer cranks through the cross-head C, side rods R, crossheads H, and connecting rods *n*. The outer cranks are at 180° to the central crank. When the pistons are moving outwards and are not far from the outer ends of their strokes, the front piston uncovers the exhaust ports E and the spent charge in the cylinder between the pistons escapes, and the pressure falls to nearly that of the atmosphere. The next event is the uncovering of the air ports A by the back piston, there is then a rush of air into the cylinder, and a thorough scavenging takes place. But before the scavenging is completed the gas ports G are uncovered and there is a

rush of gas into the cylinder, and this gas mixing with the air forms the charge for the next cycle. The pipes from the pump to the cylinder form ample reservoirs for air and gas, the former being at a gauge pressure of about 10 lb. per square inch and the latter at about 7 lb. per square inch.

During the inward strokes of the pistons the ports G, A, and E are closed and the charge between the pistons is compressed. At or near the end of the inward strokes the charge is ignited and the outward strokes of the pistons are performed under the pressure of the ignited charge in the usual way.

It will be seen that in this design of engine the moving parts practically balance one another, and the force of the explosion or the pressure on the pistons is not transmitted to the main bearings. The combustion space is of course in the centre of the cylinder between the two pistons when they are nearest together and this space is of the simplest form. It will be noticed also that for a given piston speed the expansion of the charge during the working stroke is twice as rapid in the Oechelhäuser engine as in the ordinary single piston engine, and the loss of heat to the jacket should therefore be less. There being no stuffing-boxes, no valves except a small air starting valve, and no cylinder heads, this engine is of very simple form.

The cylinder is of course water jacketed and the pistons are water cooled.

361. Stuffing-Boxes of Large Gas Engine Piston Rods.—Of internal combustion engines it is, in general, only double-acting gas

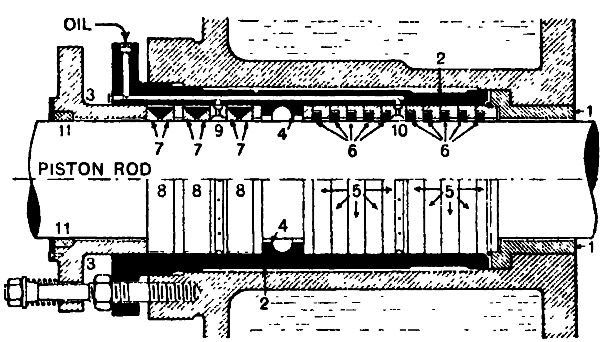


Fig. 593.

engines which require stuffing-boxes for the piston rods, and double-acting gas engines are usually of large size.

These stuffing-boxes must be well designed and should be closely watched when in use. An example of a good design is shown in Fig. 593. This stuffing-box is of the floating packing type, that is, it in no way acts as a guide or support for the piston rod.

The neck bush 1 fits into the bottom of the stuffing-box but is clear of the piston rod. The long bush 2 beds on to the neck bush with a packing ring to make a gastight joint. This long bush is flanged at its outer end and bolted to the outer part of the stuffing-box. The gland 3

fits into the bush 2 but is clear of the piston rod. The annular space between the piston rod and the bush 2 contains the packing rings. The packing space is divided into two parts by the annular projection 4 on the bush 2. The inner part of the packing space contains the L-section rings 5 which fit the bush 2 but are clear of the piston rod. Into the annular spaces formed by these rings are placed the split metal packing rings 6 which bear on the piston rod. The outer packing space contains the split white metal packing rings 7 which are backed by the rings 8. The pressure from the gland wedges the rings 7 against the piston rod. Lubrication is effected at 9 and 10 from the oil holes in the bush 2 as shown. Waste of oil is reduced by the soft packing 11.

362. Humphrey Internal - Combustion Pump.—An extremely simple and ingenious form of internal combustion engine, designed for pumping water, is due to Mr. Herbert A. Humphrey. This pump was first described by its inventor in a paper read before the Institution of Mechanical Engineers in Nov. 1909. Briefly, there is a vertical cylinder in which the cycle of operations is performed, but instead of acting on a metallic piston the gases in the cylinder act directly on the water to be pumped.

Referring to Fig. 594, which shows diagrammatically one form of the Humphrey pump, C is the cylinder or combustion chamber, over which, in the cylinder head, there are three valves the admission valve A for gas and air, the exhaust valve E, and a scavenging air valve (not shown). All these valves open inwards to the cylinder and are lightly loaded by springs. A water suction valve box V forms an extension of the cylinder. This valve box has a number of ports in it over which, on the inside, are placed valves opening inwards.

The cylinder and suction valve box are surrounded by a water supply tank or chamber W, the water to be pumped entering this tank at S. The water level in W is maintained at approximately constant level by means of a valve at S operated by a float.

A comparatively long pipe, called a *play pipe*, opens at one end into the suction valve box and at the other end into a water tower or stand pipe from which the discharge pipe D leads the water to the reservoir into which the water is to be pumped. The play pipe is shown horizontal but it is frequently placed in an inclined position rising towards the discharge pipe. As will presently be understood, the water oscillates backwards and forwards through the play pipe between the cylinder and the water tower by reason of its inertia.

At (*m*) in Fig. 594 is an indicator diagram, on a time base, given by Mr. Humphrey in a paper read before the Manchester Association of Engineers in Nov., 1910. In taking a diagram on a time base the drum of the indicator is driven by clockwork at a constant speed. It is not so convenient to take a pressure-volume diagram in this case. At (*n*) in Fig. 594 is shown approximately the form which the pressure-volume diagram would take.

The action of the pump is as follows. Commencing with the compression stroke. All the valves are closed and compression of the charge in the cylinder takes place by the column of water flowing backwards. This is shown by the curve AB at (*m*) and (*n*). The

charge is then ignited and the pressure rises to C and the stroke S_1 is completed in the time T_1. The water now descends in the cylinder under the pressure of the ignited charge which expands as shown by the expansion curve CD. At some intermediate point during the expansion of the charge the static pressure due to the head of water in the water tower will just balance the pressure of the expanding charge in the cylinder, but the water by this time has acquired considerable kinetic energy which keeps it moving even after the pressure in the cylinder has fallen to that of the atmosphere. Near the point D, at the end of the expansion, the exhaust valve opens by its own weight, the water suction valves also open and water enters from the tank W. Part of this new supply of water follows the moving water in the play

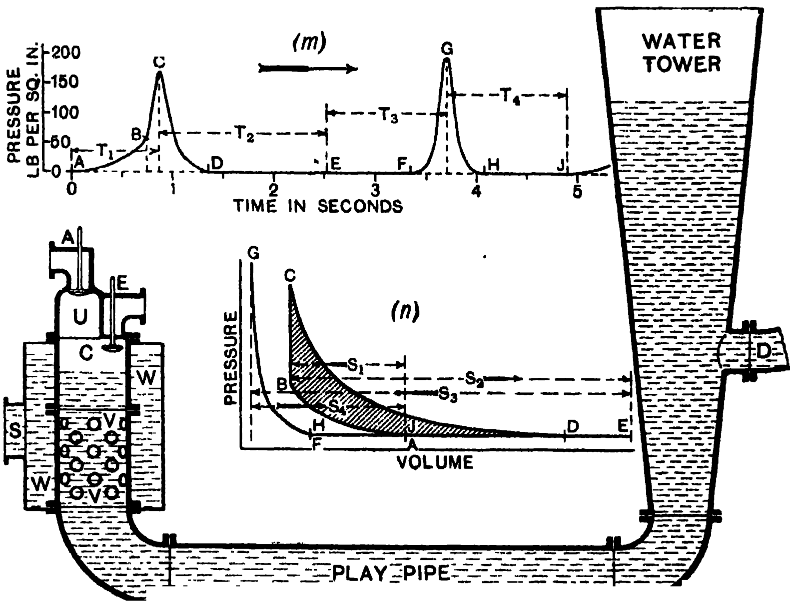


FIG. 594.—Humphrey gas pump.

pipe and there is a partial vacuum in the cylinder and the scavenging air valve opens. The remainder of the supply water rises in the cylinder and expels part of the spent charge through the exhaust valve E; the exhaust is also assisted by the scavenging air. These operations of recharging the pump with water, scavenging with air, and partial exhaust take place from D to E and the stroke S_2 is completed in the time T_2.

The water has now come to rest and is at its highest level in the water tower. The water then swings back and rising in the cylinder, the water suction valves having automatically closed, further expels the spent gases until the water impinges on the exhaust valve and closes it. The point F on the indicator diagrams shows when or where the exhaust valve closes. The scavenging valve having also closed,

and the water continuing to rise in the cylinder, the gases, mainly air from the scavenging valve, are compressed in the cushion chamber U to a pressure shown by the point G on the indicator diagrams. The water again comes to rest and the stroke S_3 is completed in the time T_3.

The compressed gases now expand and drive the water column forward until the pressure in the cylinder is again atmospheric, as shown by the point H, when the valve A opens and a charge of gas and air is drawn in. This continues until the point J is reached, which is the end of the stroke S_4 performed in the time T_4. At the end of the stroke S_4 the water again comes to rest and then swings back and another cycle starts.

The valves in the cylinder head are connected by an interlocking mechanism which prevents the scavenging and exhaust valves opening when the admission valves open and vice versa.

At (n) in Fig. 594 it will be seen that the effective work done during a cycle is represented by the hatched area ABCDA and if this be compared with the diagram of the ordinary Otto cycle it is clear that in the pump cycle the work done is greater than in the Otto cycle by the area of the toe of the diagram between A and D. For the same amount of compression the pump cycle is therefore more efficient than the Otto cycle.

There is a large installation of Humphrey pumps at Chingford, Essex. The plant consists of five pumps and four Dowson pressure gas producers using anthracite coal. Four of the pumps have cylinders 7 feet in diameter and the fifth has a cylinder 5 feet in diameter. Each of the larger pumps has sixteen exhaust valves, eight scavenging valves and eight air valves leading to eight gas valves. Full detailed illustrations of this plant will be found in *Engineering*, Feb. 14, 1913. Tests of this plant showed that each of the larger pumps was capable of delivering about 33,000 gallons of water per minute to a height of about 30 feet with an average thermal efficiency of over 22 per cent., the consumption of anthracite coal in the gas producers being under 1 lb. per pump horse-power per hour. These results prove that the Humphrey pump is not only extremely simple in construction but also very efficient.

363. Diesel Engine.—The general features of a Diesel engine are illustrated in a most instructive manner by Fig. 595, copied by permission from a diagram given by Professor W. E. Dalby in his paper, "Trials of a Small Diesel Engine," read before the Institution of Naval Architects in 1914. The diagram is not drawn to scale and the parts do not occupy their exact relative positions.

Quoting Professor Dalby's description: The central area of the diagram is occupied by the cylinder, and shows the air-admission valve, the exhaust valve, and the fuel valve which admits the supply of oil fuel to the cylinder. These valves are opened by levers operated by a half-speed cam-shaft, and are closed down on their seatings by springs. There is also a separate starting valve controlled by a separate lever and a cam on the half-speed shaft; this valve lies behind the fuel valve, but is not shown on the diagram. The supply pipe connecting the starting valve to the store bottle through the regulating valve B is, however, indicated.

To start the engine, the lever which operates the fuel valve is put temporarily out of action, so that the fuel valve keeps closed; the regulating valve B is opened, and for a few revolutions the engine runs on compressed air drawn from the store bottle. After the normal speed is reached valve B is closed, valve C is opened, and the lever controlling the fuel valve is thrown into action, oil is admitted, and the engine then works normally.

The oil supply is contained in a tank placed about 6 ft. above the

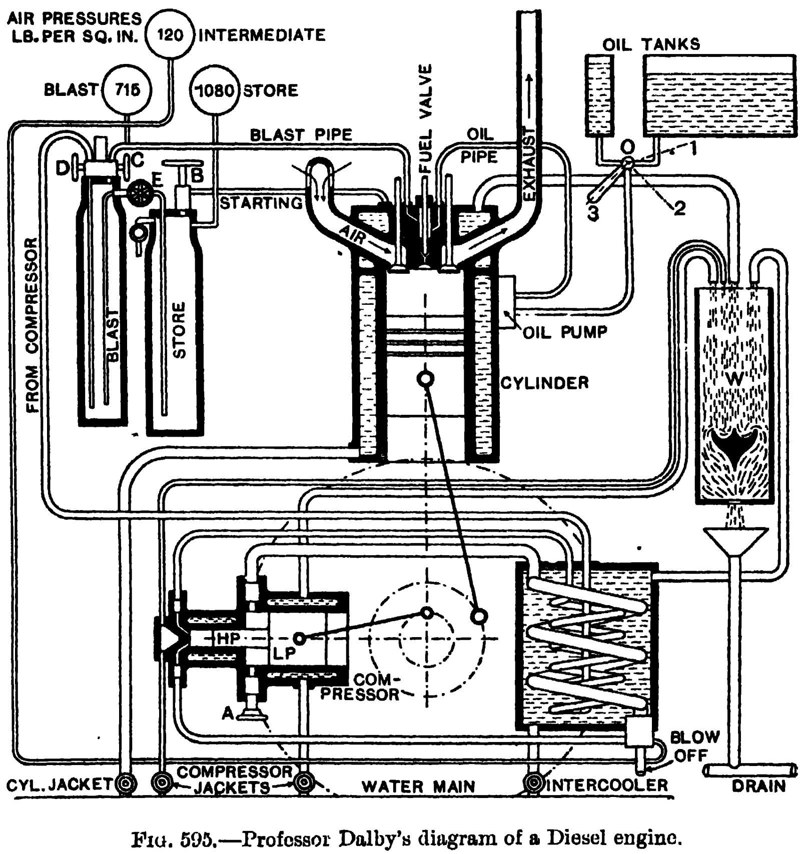


FIG. 595.—Professor Dalby's diagram of a Diesel engine.

pump. The pump forces the oil into the annular space round the fuel valve, against the blast pressure which is continuously maintained there in virtue of its connection with the blast bottle through the blast pipe and the regulating valve C. Then, when the fuel valve is opened, the blast pressure blows the oil through the narrow passage ways surrounding the nose of the fuel valve, into the clearance space of the cylinder. The difference between the pressure in the annular space and the pressure in the clearance space is considerable, and the energy corresponding to the difference is utilized in reducing the oil

to a fine mist. Air for the blast is compressed by a two-stage water-jacketed compressor driven from the crank shaft, and after each stage of the compression the air flows through a cooler, as illustrated in the diagram. Air is drawn into the compressor at A. A, in fact, is a cap by means of which the size of the opening can be varied, and in this way the quantity of air delivered to the bottles can be regulated. These bottles are placed in communication with one another through the valve E. Air is delivered to the blast bottle from the compressor

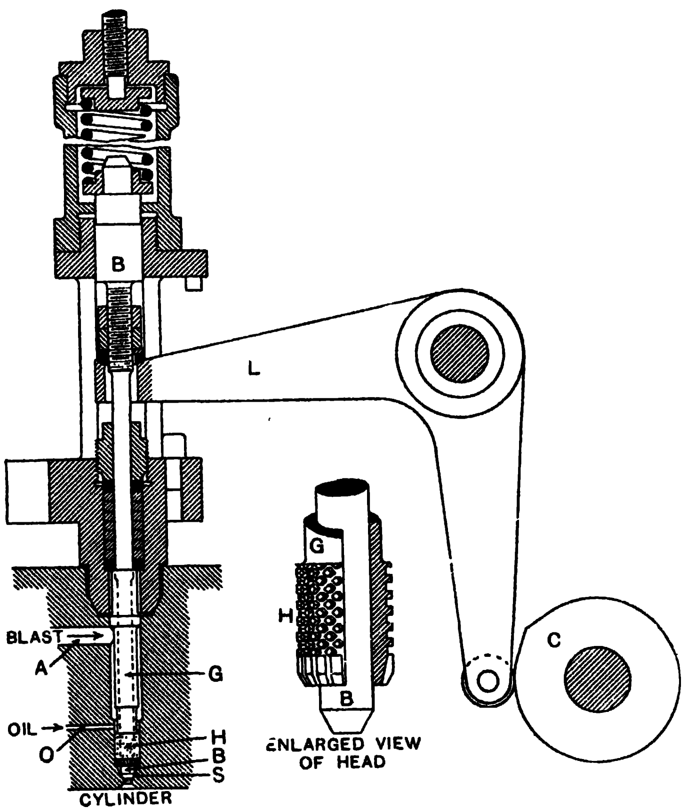


FIG. 596.—Fuel valve of Diesel engine.

through the valve D, and then, if it is also necessary to replenish the stock in the store bottle, the valve E is opened and part of the delivery flows across into store, to be used later for starting the engine; the other part maintains the supply in the blast bottle against the intermittent flow through the fuel valve. The pressure of the supply to the fuel valve is regulated by the valve C. In one of the trials made the air pressures were as noted on the diagram. Four pipes are taken from the water main to supply respectively water to the jacket of the engine cylinder, the jacket of the high-pressure cylinder of the compressor, the jacket of the low-pressure cylinder of the compressor, and

to the intercooler. The discharge from all the jackets is collected in an orifice tank W, so that the quantity of water flowing through the plant can be measured by observations of the head over the orifice.

Two oil tanks are shown in Fig. 595. The larger tank is used during the normal running of the engine while the smaller one is used during a trial. The positions 1, 2, and 3 of the handle of the three-way cock O correspond to shut off, normal running, and trial, respectively.

The fuel valve is shown in detail in Fig. 596. Quoting Professor Dalby's description: Oil from the force pump is delivered through the channel O, and air enters through the channel A. When the fuel valve is opened the air forces the oil through the narrow winding passages which obstruct its free entry into the clearance space in the cylinder. The valve itself is a steel spindle B, closing down on to the conical seating S, giving access to the clearance space of the cylinder through a hole about $\frac{1}{20}$ inch diameter. The spindle passes through a gun-metal sleeve G, the end H of which is enlarged to fill the hole just above the seating. This enlarged head is milled on the outside into narrow channels, so that the oil is atomized as it is forced through these channels on its way to the cylinder. The more usual arrangement is to form the head of the sleeve with a series of washers drilled with holes $\frac{1}{16}$ inch diameter, forming in this way passages offering high resistance to the flow of the oil. The valve spindle B is lifted by the lever L against a strong spring; and the lever L is operated by the cam C, which is placed on the half-speed shaft. The lift of the spindle B is about $\frac{3}{32}$ inch.

The fuel oil pump is shown in detail in Fig. 597. Again quoting Professor Dalby's description—S is the suction valve; D_1 and D_2 are two delivery valves in series; R is the ram. The ram is driven by a small crank and connecting rod attached to the end of the half-speed cam shaft as indicated at C by dotted lines. A small screw-down valve is provided to test the oil flow; it opens communication into the space above the first delivery valve D_1, and when open, oil spurts from the pipe P when the pump is working properly.

The quantity of oil forced through the delivery pipe during the down stroke of the ram R depends upon the fraction of the stroke of the ram at which the suction valve S closes, and this closing is determined by the action of the governor on the valve. Until the suction valve closes the ram merely displaces oil back through it into the supply chamber Y. The closing can be delayed by the action of the governor, so that, in general, part of the charge drawn in through the suction valve is turned back to the supply chamber Y through the suction valve, and the remainder is forced out through the delivery valves D_1 and D_2 to the fuel valve in the cylinder cover of the engine, against the pressure of the air in the annular chamber surrounding the fuel valve. The proportion of the whole charge which is delivered through the delivery valves during the down stroke of the ram depends upon the position of the governor sleeve E.

The mechanism by means of which the governor delays the closing of the suction valve is shown in the figure. The link H is jointed at one end I to the link L, which hangs from the arm K. The arm K is

connected to the governor sleeve, and pivots at N. The other end of the link H is connected to the crosshead pin Q, which drives the pump ram, so that as the engine works the link H executes angular oscillations about the joint I. The plug rod G is connected to the link H, so that it oscillates vertically through a range equal to about one-third of the stroke of the ram. The plug rod G passes through the pump block B, which acts as a guide, and carries below an arm J fastened to it. Assuming that the governor is rotating steadily so that the pin joint I is at rest in the position corresponding to full load, the arm J, as it oscillates up and down, keeps clear of the end of the suction valve S. If the speed increases the pin joint I moves up as the governor

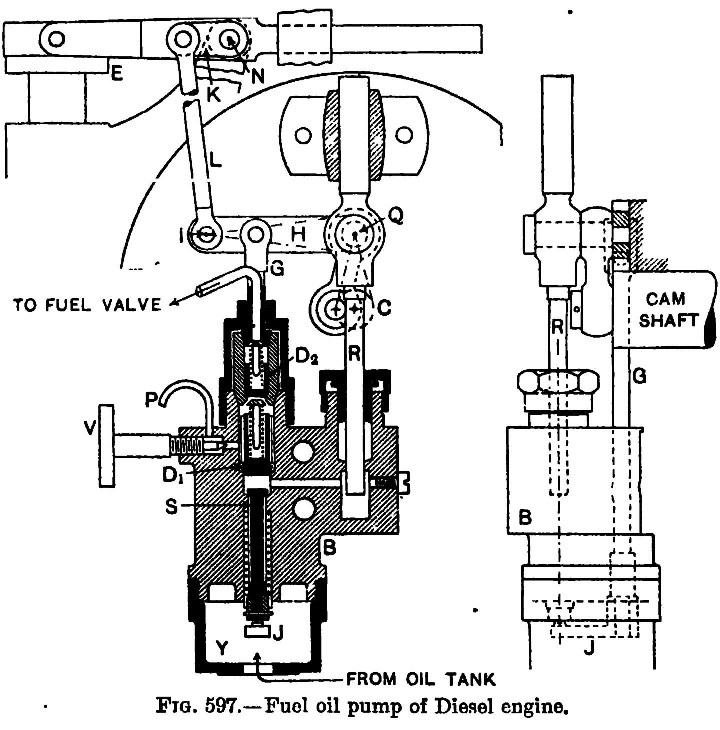


Fig. 597.—Fuel oil pump of Diesel engine.

sleeve rises and this movement brings the arm J into contact with the end of the suction valve during a portion of the stroke, and so delays its closing.

The principal dimensions of the engine are: Diameter of cylinder, $6\frac{1}{2}$ in. Stroke, $10\frac{5}{8}$ in. Air compressor H.P. cylinder, $1\frac{3}{16}$ in. diameter; L.P. cylinder, $3\frac{5}{8}$ in. diameter; stroke of pistons, $2\frac{3}{4}$ in. At full load the I.H.P. was 12·95 and the B.H.P. 9·94 at 250 revolutions per minute. For full details of the various trials which Professor Dalby made with this engine the student is referred to the paper mentioned at the beginning of this Art.

364. Campbell High Compression Oil Engine.—The illustrations, Figs. 598, 599, and 600, prepared from drawings kindly supplied by

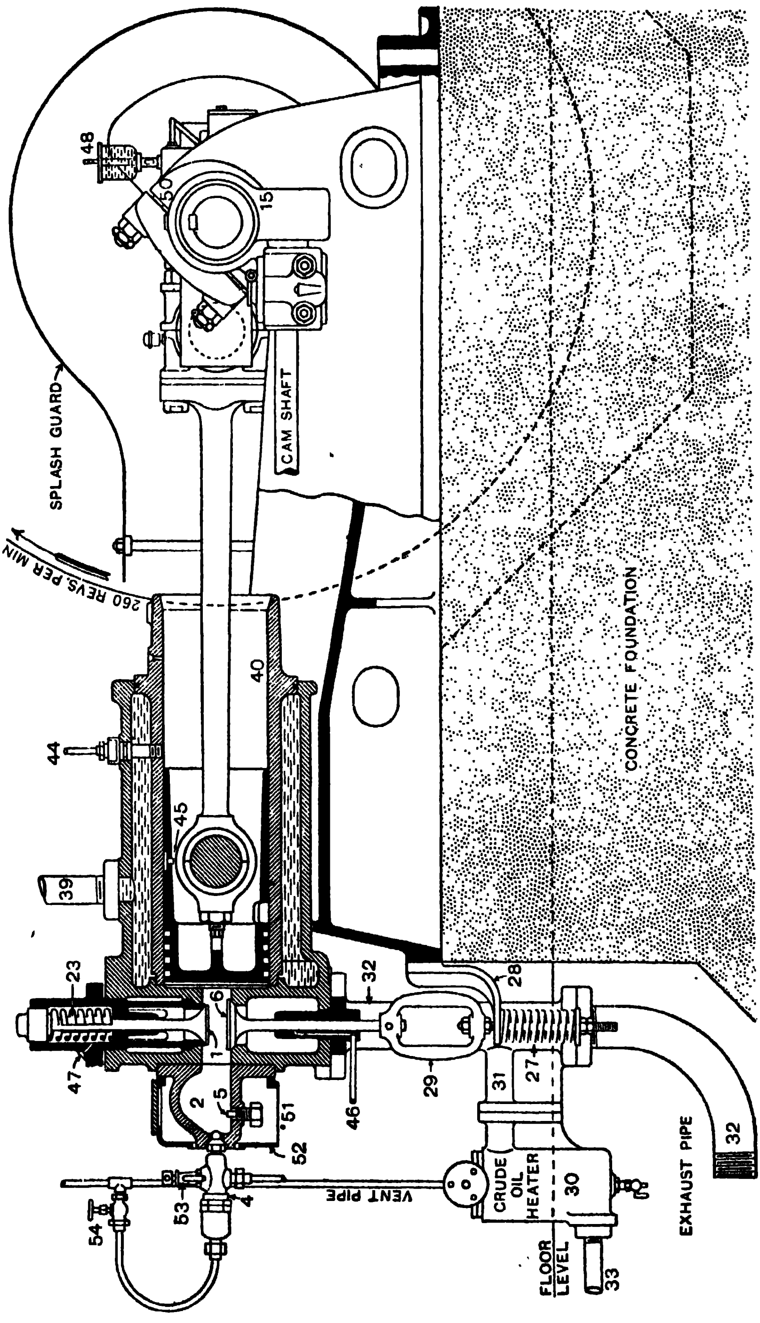


FIG. 598.—The Campbell high compression oil engine. Sectional elevation.

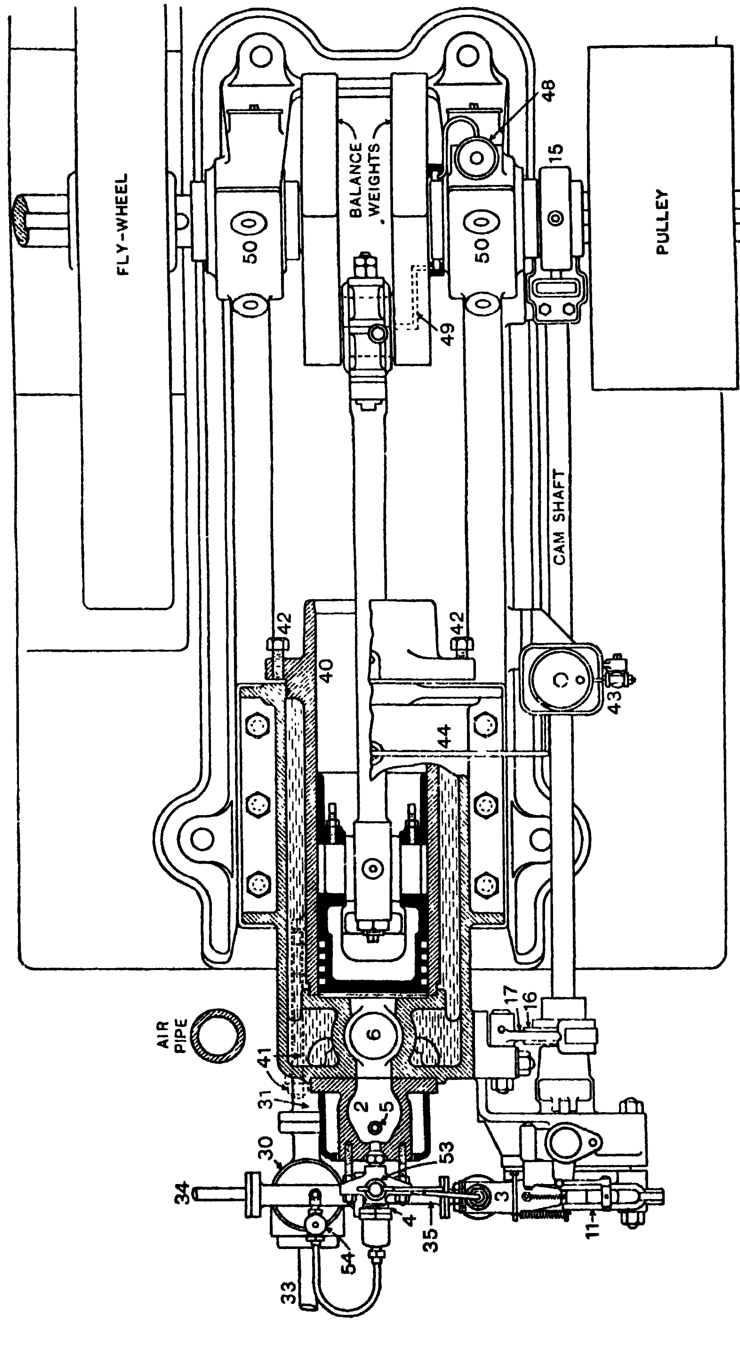


FIG. 599.—The Campbell high compression oil engine. Sectional plan.

the Campbell Gas Engine Co. of Halifax, show in considerable detail the Campbell high compression oil engine. This engine has proved highly successful with residual petroleum as fuel. This residual petroleum is frequently called crude oil, but as has already been pointed out (p. 484) the crude oil as it comes from the wells is seldom used as fuel in oil engines.

The particular engine here illustrated and described has a cylinder 7·5 inches in diameter and a stroke of 15 inches. At the normal speed of 260 revolutions per minute the working load is 11·5 horse-power and

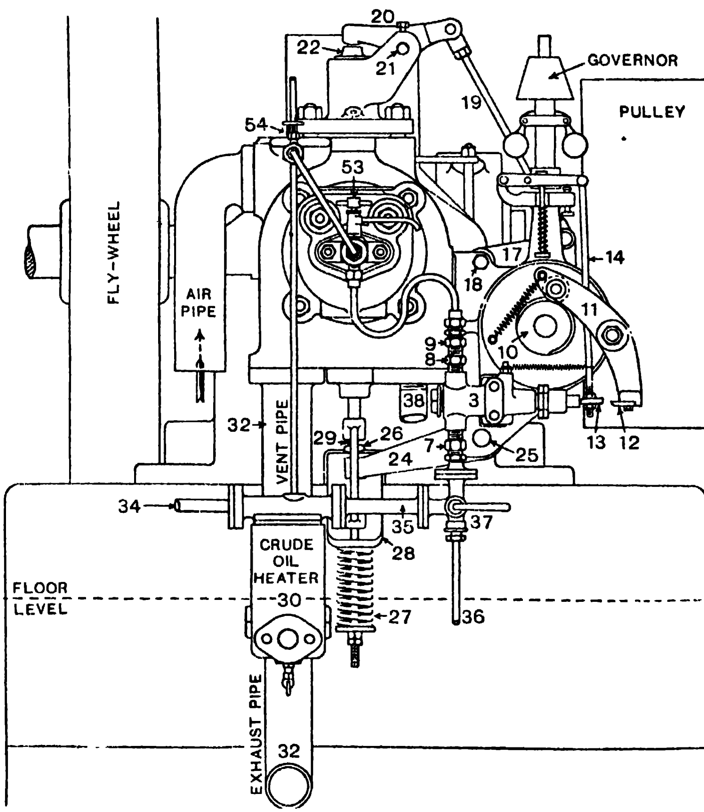


FIG. 600.—The Campbell high compression oil engine. End elevation.

the maximum load is 13 B.H.P. The engine works on the four-stroke cycle and is governed on the "hit and miss" principle.

During the suction stroke air only is drawn in through the inlet valve 1. During the next stroke this air is compressed to a pressure of about 300 lb. per square inch. Before the compression stroke is completed the fuel oil is injected into the vaporizer or hot-bulb 2 by means of the pump 3. On its way from the pump to the vaporizer the fuel oil is broken up into a fine spray or mist in the atomizer 4. The necessary temperature for the ignition of the charge is attained partly

by the heat of compression, partly by the heat of the vaporizer, and finally by the heat of the hollow ignition plug 5. Except at starting the vaporizer and ignition plug, which are not water jacketed, are heated by the combustion of the fuel oil.

On ignition there is an almost instantaneous rise in pressure to about 500 lb. per square inch. For a very short period during the succeeding working stroke the combustion is continued at approximately constant pressure as in the Diesel engine, after which there is expansion to near the end of the stroke when the exhaust valve 6 opens and exhaust begins and is completed during the fourth stroke of the cycle.

The fuel pump 3 has three ball valves, the suction valve in the plug 7, the delivery valve in the plug 8, and a check valve in the plug 9.

The in or delivery stroke of the fuel pump plunger is performed under the action of the cam 10 through the lever 11 carrying the pecker piece 12 which strikes the steel die 13 hung against the outer end of the pump plunger by the governor drop rod 14. The out or suction stroke is performed by a spring in the pump barrel. When the speed of the engine, and therefore the speed of the governor, exceeds a certain limit, due to a reduction in the load, the governor raises the die 13 clear of the pecker piece 12 and the pump is put out of action until the speed is reduced sufficiently by the temporary cutting off of the power.

The cam shaft is driven directly from the crank shaft by skew gearing in the gear case 15. The speed of the cam shaft is half that of the crank shaft.

The governor is driven by bevel gearing from the cam shaft.

The air inlet valve 1 is opened by means of the cam 16. This cam acts on a roller carried on one arm of a T-shaped radius link 17 which turns about the fulcrum pin 18. To the other arm of the radius link is jointed the rod 19 which is coupled at its upper end to one end of the lever 20. The lever 20 turns about the fulcrum pin 21, and its free end acts on the head 22 attached to the top of the air inlet valve spindle. The air inlet valve is closed by the spring 23.

The exhaust valve 6 is opened by means of the same cam 16 which opens the air inlet valve. This cam acts on the roller, not shown in the illustrations, carried on the outer end of the exhaust valve lever 24 which turns about the fulcrum pin 25. The inner end of the exhaust valve lever carries a bolt the head of which, 26, acts on the lower end of the exhaust valve spindle. The exhaust valve is closed by the spring 27. This spring abuts at its upper end against the bracket 28 and at its lower end against a washer carried by a bolt which, by means of the stirrup 29 forms an extension of the exhaust valve spindle. The inner end of the exhaust valve lever 24 works inside the stirrup 29.

The fuel oil comes from the supply tank through the pipe 34 and passes through the heater 30 and then through the pipe 35 on its way to the pump. The heater is warmed by exhaust gases from the engine cylinder which enter the jacket of the heater by the branch pipe 31 from the main exhaust pipe 32 and leave by the pipe 33. But the heater is only warmed when the fuel oil used is not sufficiently fluid to

flow freely. There is a by-pass cock (not shown) on the pipe 33 by means of which the passage of exhaust gases through the heater jacket may be cut off

For engines with what is called "cold starting" a pipe 36 is used which comes from a small secondary tank containing paraffin or gas oil. At the junction of pipes 35 and 36 there is a two-way cock 37 which can be set to pass oil from one pipe or the other to the pump. With cold starting the engine is run for a few minutes on the paraffin or gas oil and then the supply is changed over from pipe 36 to pipe 35.

The cooling water for the cylinder jacket enters by the pipe 38 and leaves by the pipe 39.

The cylinder liner 40 is secured by four T-headed bolts one of which is shown dotted at 41 in Fig. 599. The heads of these bolts lie between snugs cast on the liner. The liner may be withdrawn from the cylinder by means of the forcing screws 42.

An oil pump 43 driven from the cam shaft supplies oil through the pipe 44 to lubricate the cylinder and piston. The gudgeon pin in the piston is lubricated from the cylinder through the oil hole and short tube at 45. The exhaust valve spindle is lubricated through the oil pipe 46, and the air inlet valve spindle through the oil hole 47. The crank pin bearing in the connecting rod is supplied with oil from the lubricator 48 by the centrifugal method through the oil hole 49 in one of the crank arms. The crank shaft bearings 50 have ring lubricators running in enclosed oil baths.

Before starting the engine the vaporizer and ignition plug are heated by a blow-lamp applied at the opening 51 in the cover 52 of the vaporizer. After starting the lamp is not required, not even when the engine is running with no load.

It is important that before the first start or after disconnecting for cleaning, the fuel oil pipes should be cleared of air and charged with oil. To facilitate this the screw-down vent valve 53 is provided on the atomizer. There is also a screw-down valve 54 on the atomizer escape pipe leading to the vent pipe from the heater.

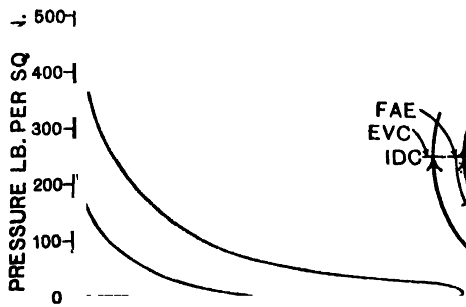


FIG. 601.

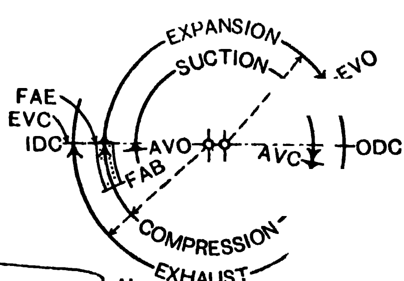


FIG. 602.

AL, atmospheric line.
AVO, air inlet valve opens.
FAE, fuel admission ends.
IDC, inner dead centre.
AVC, air inlet valve closes.
EVO, exhaust valve opens.
ODC, outer dead centre.
FAB, fuel admission begins.
EVC, exhaust valve closes.

An indicator diagram is shown in Fig. 601, and the valve setting diagram in Fig. 602.

365. Campbell Two-Stroke Cycle Oil Engine.—A good example of the two-stroke cycle oil engine with moderate compression is shown in Figs. 603 and 604. This engine is made by the Campbell Gas Engine Co. of Halifax, and the illustrations have been prepared from drawings kindly supplied by that firm.

The operations making up the two-stroke cycle have been described in Art. 351, p. 477, and it has been pointed out that in this cycle a pump for charging the cylinder with air (and also gas in the case of a

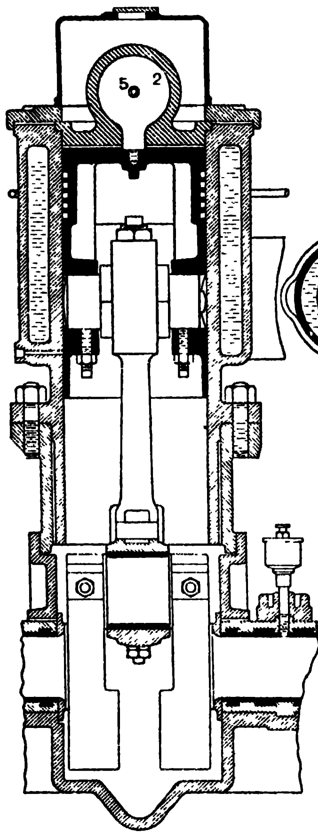


FIG. 603.

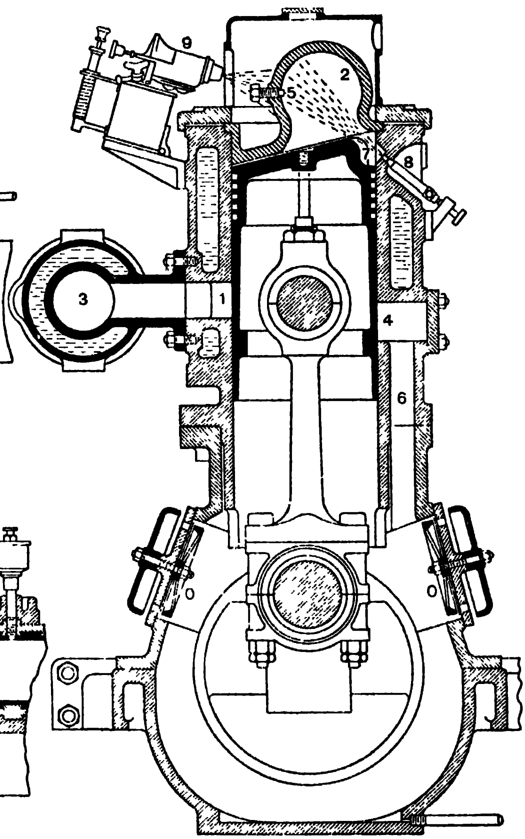
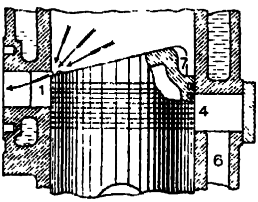


FIG. 604.

gas engine) is necessary. In the engine now to be described the engine piston and the crank case form the necessary pump. The engine is a vertical one, and during the up stroke of the piston air is drawn into the crank case through the air valves O. In these valves the moving part is a disc of sheet metal. During the succeeding down stroke the air in the crank case is slightly compressed.

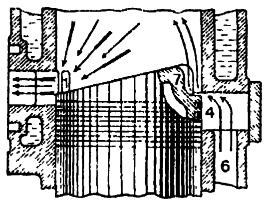
Coming now to what takes place above the piston: Not far from the end of the down stroke the piston uncovers the exhaust port 1, and the spent charge escapes through the water-jacketed exhaust pipe 3.

A little later the piston uncovers the air port 4 and the compressed air in the crank case rushes by way of the passage 6 into the cylinder. Owing to the baffle 7 formed on the piston, and to the fact that the air port opens later than the exhaust port, the inrushing air has a scavenging action, clearing out the spent gases, more or less completely, from the cylinder and vaporizer. The cylinder and vaporizer are then full of practically pure air. The operations of exhausting, scavenging, and charging the cylinder and vaporizer with air are clearly shown in Figs. 605, 606, and 607.



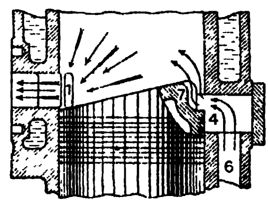
Exhaust port just open. Air port closed.

FIG. 605.



Exhaust port open. Air port just open.

FIG. 606.



Piston at bottom of stroke.

FIG. 607.

As soon as the piston covers the exhaust port during the up stroke, compression of the air above the piston begins and continues to the end of the stroke, a pressure of about 180 lb. per square inch being reached. But before the up stroke is completed the fuel oil is injected by the oil pump through the spraying nozzle 8 into the vaporizer or hot-bulb 2, where it is vaporized and the mixture of fuel vapour and air is ignited. At the top of the stroke the supply of fuel oil ceases and the down stroke is performed under the pressure of the ignited charge.

It will be seen that there is a working stroke for each revolution of the crank.

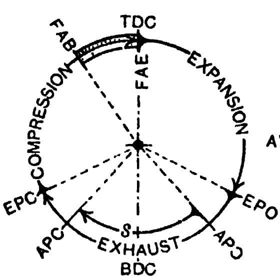


FIG. 608.

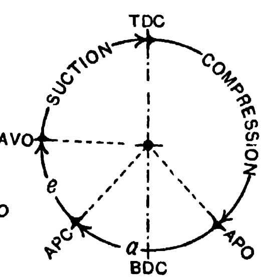


FIG. 609.

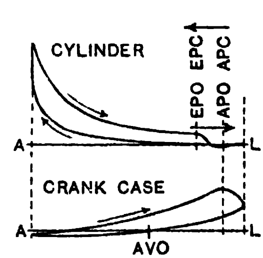


FIG. 610.

TDC, top dead centre. BDC, bottom dead centre. EPO, exhaust port opens.
EPC, exhaust port closes. APO, air port opens. APC, air port closes.
FAB, fuel admission begins (full load). FAE, fuel admission ends.
AVO, air valves in crank case open. AL, atmospheric line.
a, transfer of air from crank case to cylinder. *e*, expansion of air in crank case.
s, scavenging and charging cylinder with air from crank case.

The crank diagrams of the cycle are given in Figs. 608 and 609, the former for the cylinder above the piston and the latter for the crank case. Fig. 610 shows the form of the cylinder indicator diagram

and also the crank case indicator diagram for this type of engine, the latter being a weak spring diagram.

As in the high compression oil engine illustrated and described in Art. 364, the vaporizer 2 is heated partly by the compression of the air charge, but mainly by the combustion of the fuel oil within it. Since a charge of fuel oil is burned at every revolution of the crank the vaporizer will be more highly heated in the two-stroke cycle engine than in the four-stroke cycle engine and the ignition plug 5 is not so necessary in the former, but before starting a shorter time is required for heating up by the blow-lamp 9 when an ignition plug is provided. The blow-lamp is not required after starting.

After heating the vaporizer by means of the blow-lamp the engine is started with compressed air. This compressed air is stored in a strong cylinder or air bottle. The air bottle is generally charged with the gases from the engine cylinder, which are allowed to escape into it at the maximum pressure through the starting valve. This valve is a simple non-return valve held on its seat by a spring, which is raised by a hand lever when it is desired to admit the pressure to the cylinder for starting. Normally the valve is held down on its seat by a set screw, which of course has to be run back before the valve can be operated by hand. When it is intended to use it as a charging valve the set screw is turned back one or two turns, and this permits the valve to open slightly when the maximum pressure in the cylinder is reached and the air bottle is gradually charged. For the larger engines the same method may be adopted, but it is more general to have an independent air compressor driven in some way from the engine or by a separate small engine or motor.

The particular engine illustrated by Figs. 603 and 604 is one unit of a two-cylinder marine engine, having cylinders 11·75 inches in diameter and a stroke of 12 inches. This engine develops 66 B.H.P. as a maximum, but the full working load is 60 B.H.P. at the normal speed of 350 revolutions per minute.

366. Aero Engines.—The engines of aeroplanes and airships use petrol as fuel. They are mostly water cooled, the water being circulated by a pump and cooled in a radiator. The crank shaft speed is high, 1200 to over 2000 revolutions per minute. The engines are single acting and nearly all work on the Otto four-stroke cycle. A feature which distinguishes aero engines from all other heat engines is their extreme lightness for the power developed. The weight of the complete engine, dry, without propeller, does not generally exceed 4 lb. per B.H.P.

A part of an aero engine which differs considerably from that of other engines is the cylinder, the object being to obtain the necessary strength with the minimum of weight. Figs. 611, 612, and 613 show cylinder details of a six-cylinder 300 B.H.P. Maybach engine. Fig. 611 shows the upper part of the cylinder, Fig. 612 shows the lower part of the water jacket, and Fig. 613 shows the lower part of the cylinder barrel. Bore of cylinders, 165 mm. (6·50 in.); stroke, 180 mm. (7·09 in.).

Referring to the illustrations: H is the cast iron cylinder head, V being part of the seat of one of the four valves. The barrel B is of

steel machined all over and is only 3 mm. (0·118 in.) thick at its thinnest parts; it is screwed on to the head H as shown in Fig. 611, and is bedded on to a soft brass washer ring A. The thickness of the barrel increases from 3 mm. at the bottom of the water jacket to 4·5 mm. (0·177 in.) over the base flange F. A projection P fits into an opening in the crank case.

The water jacket J is machined all over from a cylindrical steel forging. At the upper end the jacket is screwed on to the cylinder head

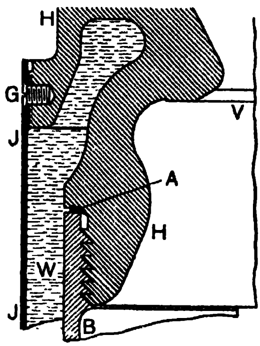


FIG. 611.

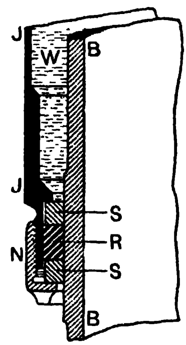


FIG. 612.

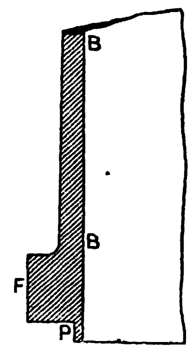


FIG. 613.

and this screwed joint is finally sweated in position with soft solder and is locked with four grub screws G. The thickness of the main part of the jacket is 1 mm. (0·039 in.). The water space W is only 7 mm. (0·276 in.) wide. The jacket extends to about two-thirds of the total length of the cylinder barrel. The joint at the bottom of the jacket is shown in Fig. 612; steel rings S have between them a soft rubber composition packing ring R compressed by means of the cap ring N screwed on to an extension of the jacket shell. This joint

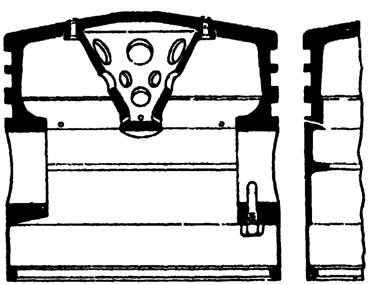


FIG. 614.

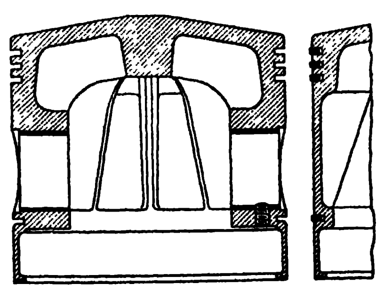


FIG. 615.

allows for the relative expansion between the cylinder barrel and the jacket.

In cheaper designs the jacket shell is made of sheet steel, the joints being acetylene-welded. Copper jacket shells are also used.

Aero engine pistons are usually made either of cast iron or of aluminium. Fig. 614 shows the Benz standard cast iron piston for

230 horse-power engines having cylinders 145 mm. (5·71 in.) in diameter. A special feature of this design is the conical steel support in the centre, riveted to the crown of the piston and bearing on the central portion of the gudgeon pin through a slot in the upper end of the connecting rod. This support causes the greater part of the load on the piston to come directly on to the centre of the gudgeon pin. This piston, complete with gudgeon pin, rings, and set screw, weighs 6·72 lb.

The Benz standard aluminium piston for the same size of engine as that just referred to is shown in Fig. 615. The head is supported and strengthened by eight webs radiating from the central boss. There are three cast iron packing rings above the gudgeon pin and one scraper ring below it. The gudgeon pin bosses are fitted with steel bushes. This piston, complete with gudgeon pin, rings, and set screw, weighs 4·91 lb.

367. Carburettors.—In engines using petrol, alcohol, and other liquid fuels which readily vaporize, the fuel is sprayed into the suction pipe of the engine by a *carburettor*. During the suction stroke air is drawn in and this air passing over a nozzle which is in communication with the fuel supply induces a current of the liquid through the mouth of the nozzle, and the liquid is sprayed into the air in minute drops which quickly vaporize.

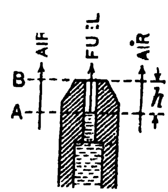


FIG. 616.

The delivery end of the nozzle is shown in Fig. 616. When not in action the fuel stands at a level A, a small distance h below the mouth of the nozzle. The fuel is fed to the nozzle from a chamber in which it is kept at the constant level A in a way to be described later.

Let v_1 = velocity of air at level B.
v_2 = velocity of the issuing fuel at level B.
w_1 = weight of 1 cubic foot of air.
w_2 = weight of 1 cubic foot of fuel.

It may be demonstrated from the theory of the flow of gases and Bernoulli's theorem that at the mouth of the jet, friction being neglected, $\frac{w_2 v_2^2}{2g} + w_2 h = \frac{w_1 v_1^2}{2g}$

and $v_2 = \sqrt{\frac{w_1 v_1^2}{w_2} - 2gh}$

The velocity of the air at which the flow of fuel ceases, or $v_2 = 0$, is given by, $\frac{w_1 v_1^2}{w_2} = 2gh$, or $v_1 = \sqrt{\frac{2ghw_2}{w_1}}$

For example, taking $h = 0·15$ inch or 0·0125 foot, and $\frac{w_2}{w_1} = 600$, the critical velocity of the air is $\sqrt{2 \times 32·2 \times 0·0125 \times 600} = 22$ feet per second.

Once the flow of fuel is started it will be found from the formula that the fuel velocity increases more rapidly than the air velocity, and this is verified in practice. This is best shown by numerical examples.

First take $v_1 = 60$ feet per second, then, taking the same values of

h and $\frac{w_2}{w_1}$ as before, $v_2 = \sqrt{\frac{60^2}{600} - 2 \times 32 \cdot 2 \times 0 \cdot 0125} = 2 \cdot 28$ feet per second.

Next let the air velocity be doubled, or $v_1 = 120$ feet per second, then $v_2 = \sqrt{\frac{120^2}{600} - 2 \times 32 \cdot 2 \times 0 \cdot 0125} = 4 \cdot 82$ feet per second, which is 2·11 times the former fuel velocity.

This increase in the relative velocity of fuel and air, as the velocity of the air increases, is greater the greater the value of h. If h is zero, then $\frac{v_2}{v_1} = \sqrt{\frac{w_1}{w_2}}$, which is constant for the same fuel at the same temperature of fuel and air. It is, however, necessary that h shall have a positive value to prevent loss of fuel when the engine is not running.

For a given fuel there is a particular mixture of air and fuel which gives the best results. Now, if for a particular speed of air the proportions of the fuel nozzle and air passage give the correct mixture of air and fuel, it follows that at other air speeds the mixture will be different, since the amounts of air and fuel passing are proportional to their respective velocities. This complicates the problem of carburettor design, and the numerous designs of carburettors on the market is one of the consequences.

A carburettor which is very extensively used is the *Zenith carburettor*, which will now be described.

Referring to Fig. 617, which is a vertical section of the most common form of the Zenith carburettor, the fuel from the main supply tank enters at 1 and passes into the float chamber 2, the purpose of which is to keep the fuel at an approximately constant level. The constant level in the float chamber is maintained by means of the float 3 and needle valve 4 operated through the levers 5. The needle 6 of the needle valve has a grooved collar 7 attached to it, and the inner ends of the levers 5 work in the groove of this collar. The levers 5 turn on pins 8. The outer ends of the levers, which are enlarged to form counterweights, rest on the top of the float.

If the fuel level is below the normal the needle valve is open and the fuel enters the float chamber at a greater rate than the engine requires, the float rises and the needle is depressed and when the normal fuel level is reached the needle valve closes and the fuel supply to the float chamber is cut off. When the fuel level again falls below the normal the fall of the float opens the needle valve.

The carburettor is connected to the engine suction pipe at 9. The air supply enters at 10 under the control of the throttle valve 11, which is attached to the spindle 12, which is operated by means of levers outside the carburettor. The passage below the throttle valve is contracted by the choke tube 13 for the purpose of increasing the velocity of the air in the neighbourhood of the fuel nozzles 14 and 15. These nozzles form a compound nozzle, of which the inner part 14 is called the main nozzle and the outer part 15 is called the compensating nozzle. The outlet from the compensating nozzle is annular. The main nozzle 14 has free communication with the float chamber.

The compensating nozzle also communicates with the float chamber, but the supply is limited by having to pass through the small inlet 16. Over the inlet 16 there is a chamber 17 which is in communication with the atmosphere through the air hole 18.

The part of the fuel supply to the engine which passes through the main nozzle 14 will tend to increase the richness of the mixture as the speed of the engine, and therefore the speed of the air, increases; but owing to the fact that the chamber 17 over the inlet 16 is in free

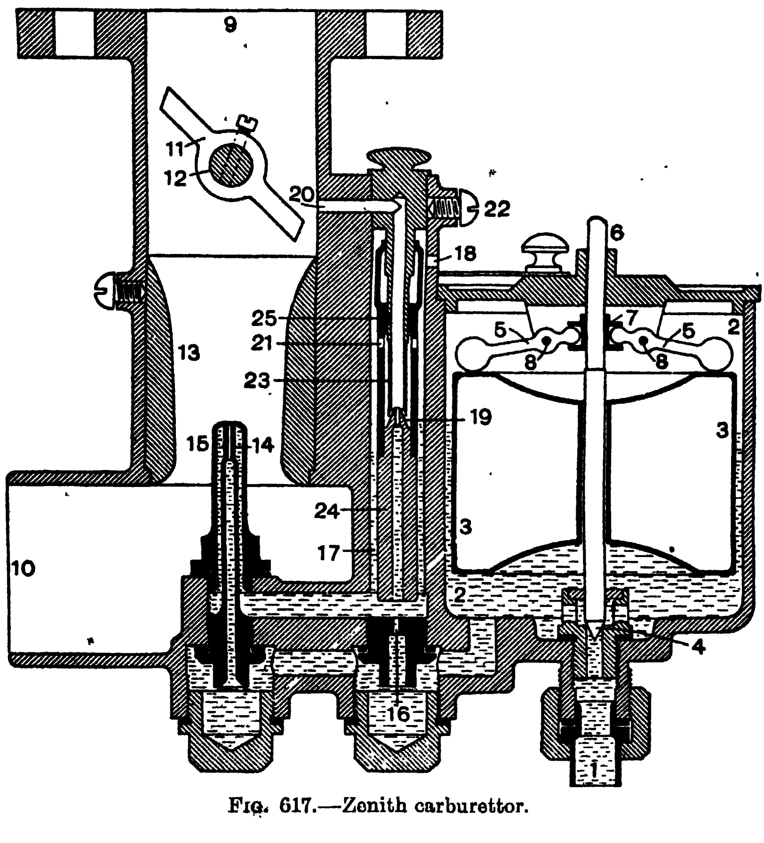


FIG. 617.—Zenith carburettor.

communication with the atmosphere through the air-hole 18, the part of the fuel supply passing through the compensating nozzle will not increase, and the part of the fuel supply to the engine which passes through the compensating nozzle will tend to decrease the richness of the mixture as the speed of the engine increases. The consequence is that, if the parts are properly proportioned, the combined supply by the two nozzles 14 and 15 will always bear the same ratio to the air supply.

A separate nozzle 19 comes automatically into action for slow running or for starting when the throttle valve 11 is only slightly

open. In that case the air velocity in the choke tube is not sufficient to operate the nozzles 14 and 15, but in the contracted passage round the slightly open throttle valve there is sufficient air velocity to operate the nozzle 19 through the by-pass 20 near the edge of the throttle valve. When the throttle valve is open a certain amount, the air velocity at the mouth of the by-pass 20 is not sufficient to operate the nozzle 19, and it automatically goes out of action.

There is an air supply to the nozzle 19 through the holes 21, and this supply can be adjusted so that any strength of mixture may be arranged for starting purposes To adjust this air supply the set screw 22 is slackened back, and the whole of the slow running and starting device is lifted out and the two parts 23 and 24 are then screwed nearer together or further apart, these parts having a screw connection at 25

368. Four-Stroke Cycle *versus* Two-Stroke Cycle Engines.—The relative merits of the four-stroke cycle and the two-stroke cycle for internal combustion engines have given rise to much discussion, particularly in the application of the Diesel engine to the propulsion of ships, and the views expressed by experienced engineers and engine builders have been somewhat conflicting.

Considering single-cylinder single-acting engines, the two-stroke cycle engine has twice as many working strokes as the four-stroke engine for the same number of revolutions of the crank shaft. The first consequence of this is that the turning effort on the crank shaft is much more uniform in the case of the two-stroke cycle engine, which is a great advantage The second consequence is that but for the shortening of the effective stroke by the exhaust ports in the two-stroke cycle engine, and the addition of the pump for scavenging and charging the cylinder with air, also in the case of gas engines the addition of a gas pump, the power of the two-stroke cycle engine would be double that of the other. In practice, however, the extra power of the two-stroke cycle engine is only from 70 to 90 per cent. instead of 100 per cent

In the four-stroke cycle engine the pumping is performed in the engine cylinder, while in the two-stroke cycle engine it is performed in a separate cylinder, which has to carry a pressure of only 4 or 5 lb. per square inch and can readily be made double-acting. Notwithstanding the presence of the extra cylinder or cylinders for pumping in the case of the two-stroke cycle engine, this engine is lighter than the four-stroke cycle engine of the same power. The more uniform turning effort on the crank shaft of the two-stroke cycle engine also means a lighter fly-wheel

In the past the four-stroke cycle engine has been more economical in fuel than the two-stroke cycle engine, but in recent years, with increasing experience in designing and operating the latter type, its economy in fuel has been much improved and is closely approaching that of the former type.

For the same crank shaft speed the valve gear, and in an oil engine the fuel pump, of a two-stroke cycle engine work at twice the speed required in a four-stroke cycle engine. This means greater noise and wear and tear in the two-stroke cycle engine.

The gear for reversing a two-stroke cycle engine is simpler than that for the four-stroke cycle engine.

The absence of an exhaust valve in the two-stroke cycle engine having exhaust ports in the wall of the cylinder is a great advantage in large engines.

In the case of two-stroke cycle gas engines there is liability to the loss of gas through the exhaust ports in scavenging and charging the cylinder, but this does not apply to oil engines in which air only is introduced while exhaust is taking place. Large gas engines generally use a cheap gas and a slight waste through the exhaust ports is not of great importance.

For stationary engines of small and medium powers the four-stroke cycle is generally used on account of its simplicity and proved economy, and even for large power stationary engines it is much used. For marine purposes, however, the advantages of reduction of weight and space occupied are greatly in favour of the two-stroke cycle type of engine.

369. Combined Internal Combustion and Steam Engine.—Owing to the high temperature produced in the cylinder of an internal combustion engine, a partial cooling of the cylinder is a necessity, and this cooling means a waste of heat. In a steam engine the steam loses heat to the cylinder, but the cylinder is never too hot and no external cooling is necessary, in fact the steam in the cylinder would work more efficiently if the cylinder could be kept hotter than the steam. These and other considerations have caused suggestions and experiments to be frequently made in the past to combine the internal combustion engine and the steam engine in one prime mover. What promises to be a successful engine of this type is that invented by Mr. W. J. Still. This engine was first publicly described in a paper by Mr. F. E. D. Acland, read before the Royal Society of Arts in May, 1919.

One form of the Still engine is shown diagrammatically in Fig. 618, prepared from one given in Mr. Acland's paper. Referring to Fig. 618, AB is a cylinder in which works the piston C. An internal combustion engine cycle is performed in the cylinder above the piston and a steam engine cycle below the piston. As arranged in Fig. 618, the internal combustion cycle is a two-stroke one. It will be seen that the piston has a length rather greater than the stroke.

The upper part of the cylinder is water jacketed, while the lower part is steam jacketed. A peculiar feature of the water jacket is that the water in it is kept at a practically constant temperature, and its cooling action is due to part of the water being evaporated. The heat transferred to the water jacket is therefore represented by the latent heat of the steam produced, and the temperature of the water corresponds to the pressure of the steam.

The steam produced in the jacket passes to the boiler EF together with a portion of the water.

The exhaust gases from the combustion end of the cylinder pass first through the tubular heater GH and then through another heater HK before passing to the atmosphere. The heater HK warms the feed water which joins the circulating water from the lower part of

the boiler at the bottom of the heater GH. The water passing through the heater GH forms the supply to the water jacket.

Steam passes from the boiler through the pipes shown to the steam jacket and then to the underside of the piston C. The admission and

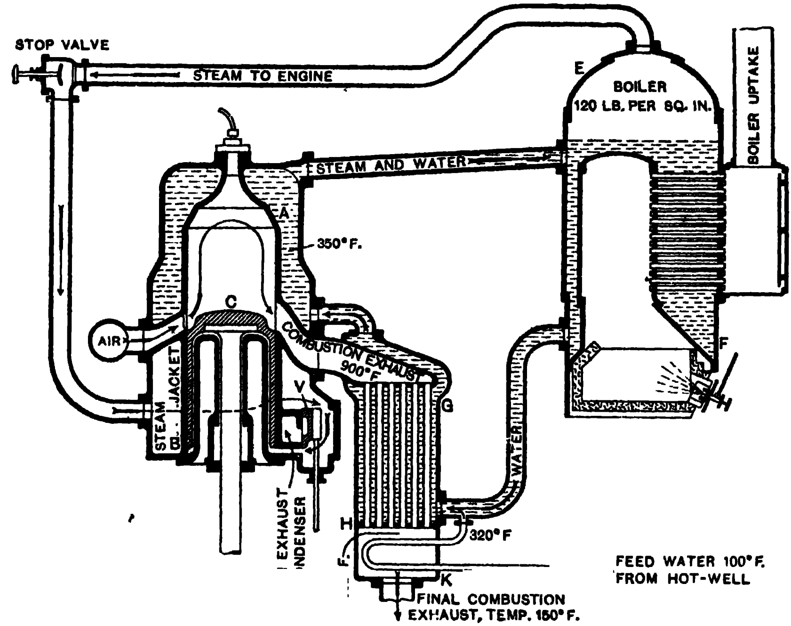


FIG. 618. —The Still combined internal combustion and steam engine.

discharge on the underside of the piston are controlled by the valve V worked from the crank shaft.

Steam for the first starting of the engine is raised by burning oil or other fuel in the fire-box of the boiler. Also, in this way, extra steam for overload may be produced, but it must be noted that in obtaining extra power in this way the total efficiency diminishes, because the fuel is much less efficient in the boiler than in the internal combustion engine.

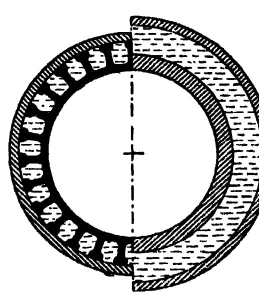


FIG. 619.

A special feature of the Still engine is the construction of the cylinder, a half cross section of which is shown to the left in Fig. 619. For comparison the right-hand half of Fig. 619 shows the ordinary construction of internal combustion engine cylinders. The Still cylinder consists of an inner cast iron liner, which is approximately one-third to one-fourth of the usual thickness; it is ribbed externally so as to add to its conducting surface and provide suitable passages for the cooling water, and it is reinforced by an outer steel shell capable of withstanding the highest pressure to

be met with in working. It seems to the writer that this construction of cylinder might be used with advantage for large internal combustion engines other than the Still engine; also there might be a further advantage in arranging the ribs helically.

The following advantages are claimed for the Still engine:—

The higher mean temperature of the engine makes the combustion cycle more effective and more reliable in starting and in running.

The mechanical stresses due to heat are reduced.

The steam generated by the heat rejected by the combustion cycle, (*a*) augments the power of the engine without increasing the consumption of fuel, (*b*) cools the piston, (*c*) reduces the mechanical losses of the engine, (*d*) is used under exceptionally favourable conditions, since it absorbs heat during expansion, (*e*) suffers no condensation loss in the working cylinder, (*f*) warms the engine before starting, so that no sudden temperature changes occur in any part, (*g*) starts the engine against load if necessary, (*h*) enables manœuvring to be performed at any speed from nothing to maximum and for any period.

Power at normal loads is developed by combustion and steam from waste heat alone, with an efficiency at least 25 per cent. above any known combustion engine of similar proportions.

Since the steam raised by the waste heat is more than enough to overcome the mechanical losses of the engine, the B.H.P. of the engine is at least equal to the I.H.P. produced by the combustion.

Additional steam gives a large range of overload without serious loss in efficiency.

In engines using heavy oil as fuel, the compression pressures which ensure ignition can be reduced by 50 per cent., as compared with the Diesel system.

For a record of tests of a Still engine see Art. 400, p. 565.

Exercises XXII

1. A cubic foot of methane or marsh gas (CH_4) weighs 0·0448 lb. at standard pressure and temperature. The higher calorific value of this gas is 13344 C.H.U. per lb. Compute the higher and lower calorific values of a cubic foot of this gas, taking the latent heat of steam at standard pressure as 539·2 C.H.U. per lb.

2. A town gas which has a lower calorific value of 312 C.H.U. per cubic foot requires 5·36 cubic feet of air for its complete combustion. A blast furnace gas which has a lower calorific value of 56·1 C.H.U. per cubic foot requires 0·71 cubic foot of air for its complete combustion. Compute for each of these gases the lower calorific value of a cubic foot of mixture containing just sufficient air for complete combustion.

3. The gas from a certain suction gas producer, using bituminous coal, had the following composition, by volume, per cent.: CH_4, 5·4; H_2, 9·5; CO, 15·4; CO_2, 7·8; N_2, 61·9. Determine the higher and lower calorific values of this gas in C.H.U. or B.Th.U. per cubic foot.

4. The composition of the gas from a suction gas producer using coke was found to be, by volume, per cent., as follows: CH_4, 0·4; H_2, 12·0; CO, 25·5; CO_2, 5·3; O_2, 0·65: N_2, 56·15. Calculate the higher and lower calorific values of this gas per cubic foot and per lb. Determine also the minimum volume of air required for the complete combustion of 1 cubic foot of this gas.

5. A particular brand of petrol having a specific gravity of 0·70 and a calorific value of 10,600 C.H.U. per lb. is sold at 3*s*. per gallon. What should be the price per gallon of another brand of petrol having a specific gravity of 0·75 and a calorific value of 10,200 C.H.U. in order that the cost per unit of heat energy may be the same for both brands?

CHAPTER XXIII

THEORY OF INTERNAL COMBUSTION ENGINES

370. Efficiency of the Otto Cycle.—The Otto cycle has been described in Art. 349, p. 475. The PV diagram for the ideal Otto four-stroke cycle is shown in Fig. 620. The following assumptions will be made. (1) During the suction stroke EA and during the exhaust stroke AE the pressure in the cylinder is atmospheric. (2) The compression curve AB and the expansion curve CD are adiabatics whose equations are of the form PV^n = constant. (3) All the heat received by the gases is received while they are at the constant volume V_B and while the pressure rises from B to C, that is during explosion. Also all the heat rejected is rejected at the constant volume V_A at the end of the

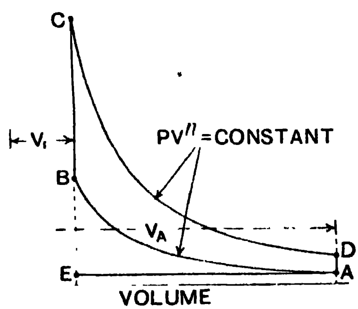


FIG. 620

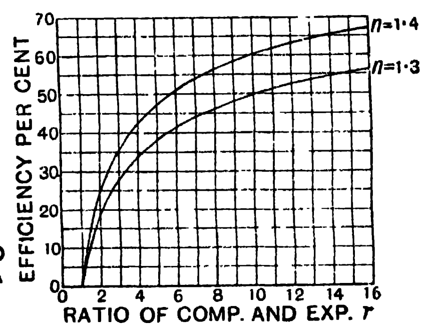


FIG. 621.

working stroke while the pressure falls from D to A. (4) The specific heats of the gases are constant.

Let P_A, P_B, P_C, and P_D, denote the absolute pressures at the points A, B, C, and D respectively, V_A, V_B, V_C, and V_D the corresponding volumes, and T_A, T_B, T_C, and T_D the corresponding absolute temperatures. Consider unit weight of gases.

Heat received between B and C $= K_v(T_C - T_B)$.

Heat rejected between D and A $= K_v(T_D - T_A)$.

Heat converted into work $= K_v(T_C - T_B) - K_v(T_D - T_A)$.

$$\text{Thermal efficiency} = \frac{\text{Heat converted into work}}{\text{Heat received}}$$

$$= \frac{K_v(T_C - T_B) - K_v(T_D - T_A)}{K_v(T_C - T_B)}$$

$$= 1 - \frac{T_D - T_A}{T_C - T_B}.$$

But $\frac{T_D}{T_C} = \frac{P_D V_D}{P_C V_C}$ and $\frac{P_D}{P_C} = \left(\frac{V_C}{V_D}\right)^n$, therefore $\frac{T_D}{T_C} = \left(\frac{V_C}{V_D}\right)^{n-1} = \left(\frac{1}{r}\right)^{n-1}$
where r is the ratio of expansion and also the ratio of compression.

In like manner it may be shown that $\frac{T_A}{T_B} = \left(\frac{1}{r}\right)^{n-1}$

Therefore $\frac{T_D}{T_C} = \frac{T_A}{T_B}$, and, by a well known theorem in algebra, each of these is equal to $\frac{T_D - T_A}{T_C - T_B}$

Hence the thermal efficiency is equal to $1 - \left(\frac{1}{r}\right)^{n-1}$

This shows that the thermal efficiency is greater the greater the value of r. The following table gives values of the thermal efficiency in percentages for various values of r and n.

Thermal Efficiency of Otto Cycle. Per Cent.

1·40	35·6	42·6	47·5	51·2	54·1	56·5	60·2	66·1
1·35	31·9	38·4	43·1	46·6	49·4	51·7	55·3	61·2
1·30	28·1	34·0	38·3	41·6	44 2	46·4	49·9	55·6

The relation of the efficiency to r, for $n = 1·4$, and $n = 1·3$ is also shown by the curves in Fig. 621.

371. Air Standard Efficiency.—Of the gases forming the charge in an internal combustion engine, air forms by far the largest part and if the volume of the combustible gases be neglected and the whole charge be assumed to consist of air, then the value of the index n in the expression $1 - \left(\frac{1}{r}\right)^{n-1}$ of the preceding Art. becomes 1·4 and the efficiency $1 - \left(\frac{1}{r}\right)^{0·4}$ is called the *air standard efficiency*.

372. Relative Efficiency.—Knowing the indicated work, and the quantity of fuel supplied per cycle, and its calorific value, the actual thermal efficiency (heat converted into work divided by heat supplied) may be computed, and if this is expressed as a fraction of the air standard efficiency for the same ratio of compression r as in the actual engine, then the result is called the *relative efficiency* of the engine.

For example, if in a gas engine in which the compression ratio is 5, the quantity of gas supplied per cycle is 0·078 c. ft. and its calorific value is 302 C.H.U. per c. ft. and if the work shown by the indicator diagram is 10,553 ft.-lb. per cycle, then the heat converted into work is $\frac{10,553}{1400}$ C.H.U. The heat supplied is 0·078 × 302 C.H.U. and the

actual thermal efficiency of the engine is $\dfrac{10{,}553}{1400 \times 0{\cdot}078 \times 302} = 0{\cdot}320$.

The air standard efficiency is $1 - \left(\dfrac{1}{5}\right)^{0{\cdot}4} = 0{\cdot}475$, and the relative

efficiency is $\dfrac{0{\cdot}320}{0{\cdot}475} = 0{\cdot}674$ or $67{\cdot}4$ per cent.

Sometimes the term "efficiency ratio" is improperly used instead of "relative efficiency" as defined above. The approved definition of efficiency ratio is given in the next Art.

373. Efficiency Ratio.—If an ideal efficiency be computed assuming that the actual engine mixture is used with the same compression ratio and that allowance is made for the variation in the specific heats with change of temperature, then the ratio of the actual efficiency to this ideal efficiency is called the *efficiency ratio.* The method of computing the ideal efficiency referred to is explained in Art. 383, p. 530.

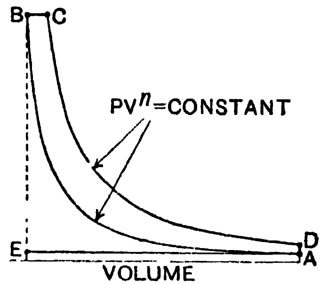


Fig. 622.

374. Efficiency of the Diesel Cycle.—The Diesel cycle has been described in Art. 350, p. 476. The PV diagram for the ideal Diesel four-stroke cycle is shown in Fig. 622. The notation used in Art. 370 on the efficiency of the Otto cycle will be used here and the same assumptions will be made.

Heat received between B and C $= K_p(T_C - T_B)$.
Heat rejected between D and A $= K_v(T_D - T_A)$.
Heat converted into work $= K_p(T_C - T_B) - K_v(T_D - T_A)$.
Thermal efficiency

$$= \frac{K_p(T_C - T_B) - K_v(T_D - T_A)}{K_p(T_C - T_B)} = 1 - \frac{K_v}{K_p}\left(\frac{T_D - T_A}{T_C - T_B}\right)$$

Let r_1 denote the compression ratio $\dfrac{V_A}{V_B}$ and r_2 the expansion ratio $\dfrac{V_D}{V_C} = \dfrac{V_A}{V_C}$

$$T_B = T_A\left(\frac{V_A}{V_B}\right)^{n-1} = T_A r_1^{\,n-1} \qquad \frac{T_C}{T_B} = \frac{V_C}{V_B} = \frac{V_A}{V_B} \div \frac{V_A}{V_C} = \frac{r_1}{r_2}$$

Hence, $T_C = T_B\dfrac{r_1}{r_2} = T_A\dfrac{r_1^{\,n}}{r_2}$; $\quad T_D = T_C\left(\dfrac{V_C}{V_D}\right)^{n-1} = \dfrac{T_C}{r_2^{\,n-1}} = T_A\left(\dfrac{r_1}{r_2}\right)^n$

Therefore,

$$\frac{T_D - T_A}{T_C - T_B} = \frac{T_A\left(\frac{r_1}{r_2}\right)^n - T_A}{T_A\frac{r_1^{\,n}}{r_2} - T_A r_1^{\,n-1}} = \frac{\left(\frac{r_1}{r_2}\right)^n - 1}{r_1^{\,n-1}\left(\frac{r_1}{r_2} - 1\right)}$$

Hence the expression for the thermal efficiency may be put in the form

$$\text{Thermal efficiency} = 1 - \frac{1}{n}\,\frac{\left(\frac{r_1}{r_2}\right)^n -}{r_1^{\,n-1}\left(\frac{r_1}{r_2} - 1\right)}$$

Putting $n = 1 \cdot 4$, the above expression gives the air standard efficiency for the Diesel cycle.

375. Comparison of Otto and Diesel Cycles.—In all applications hitherto made of the Otto cycle, the fuel gas or vapour has been mixed with the air before compression and this has limited the amount of compression because of the danger of pre-ignition due to the heating of the mixture by compression, and it has been shown that the greater the compression the greater is the efficiency. By compressing the air separately as is done in the Diesel cycle and afterwards spraying the oil fuel into the compressed air all danger of pre-ignition is avoided and consequently the advantage of high compression may be obtained, but the full advantage of the high compression cannot be reaped because, as practised in the Diesel cycle, the fuel is burned at practically constant pressure which is a less efficient process than burning it at constant volume as in the Otto cycle. This will be best shown by the following examples.

Four cases will be considered in each of which the same quantity of fuel will be assumed to be burned, or the same quantity of heat supplied, and the same quantity of air will be assumed to be used. The calculations will be simplified by neglecting the volume occupied by the fuel, which is small compared with the volume of the air. It will be assumed in each case that the temperature of the air at the beginning of the compression is 100° C. or 373° C. absolute.

All pressures are in lb. per square inch absolute and temperatures are in degrees Centigrade absolute.

The equations for the adiabatics will be taken as $PV^{1 \cdot 4} = \text{constant}$.

Case I. *Otto cycle.* (Fig. 623).—$P_A = 15$. $P_B = 200$. $P_C = 500$.

$$\frac{P_D}{P_A} = \frac{P_C}{P_B} = \frac{500}{200}, \text{ therefore } P_D = \frac{15 \times 500}{200} = 37 \cdot 5.$$

$$P_B V_B^{1 \cdot 4} = P_A V_A^{1 \cdot 4}, \text{ therefore } V_B = V_A\left(\frac{P_A}{P_B}\right)^{\frac{1}{1 \cdot 4}} = 0 \cdot 1572 V_A.$$

$$T_A = 373. \qquad T_B = T_A \frac{P_B V_B}{P_A V_A} = 781 \cdot 8.$$

$$T_C = T_A \frac{P_C V_C}{P_A V_A} = 1954 \cdot 5. \quad T_D = T_A \frac{P_D V_D}{P_A V_A} = 932 \cdot 5.$$

Heat received between B and C $= (1954 \cdot 5 - 781 \cdot 8)K_v = 1172 \cdot 7\, K_v$.
Heat rejected between D and A $= (932 \cdot 5 - 373)\, K_v = 559 \cdot 5\, K_v$.
Heat converted into work $= (1172 \cdot 7 - 559 \cdot 5)\, K_v = 613 \cdot 2\, K_v$.

$$\text{Thermal efficiency} = \frac{613 \cdot 2 K_v}{1172 \cdot 7 K_v} = 0 \cdot 523 \text{ or } 52 \cdot 3 \text{ per cent.}$$

Mean effective pressure (M.E.P.)

$$\frac{\dfrac{P_CV_C - P_DV_D}{n-1} - \dfrac{P_BV_B - P_AV_A}{n-1}}{V_A - V_B} = 73{\cdot}1.$$

CASE II. *Diesel cycle.* (Fig. 623).—The compression curve AB is the same as in Case I. At the end of the compression heat is supplied at the constant pressure 200 while the volume is increased by the amount BC (dotted line). The dotted curve CD is the expansion line.

As in Case I, $T_B = 781{\cdot}8$. The heat supplied between B and C is the same as in Case I, hence, $(T_C - 781{\cdot}8)K_p = 1172{\cdot}7K_v$.

Therefore $\qquad T_C = \dfrac{1172{\cdot}7}{1{\cdot}4} + 781{\cdot}8 = 1619{\cdot}4.$

As in Case I, $V_B = 0{\cdot}1572V_A$. $\qquad V_C = V_B\dfrac{T_C}{T_B} = 0{\cdot}3256V_A.$

$P_DV_D^{1{\cdot}4} = P_CV_C^{1{\cdot}4}.$ $\qquad$ Hence $P_D = 41{\cdot}57.$

$\dfrac{T_D}{T_A} = \dfrac{P_D}{P_A}$, hence $T_D = 1033{\cdot}7.$

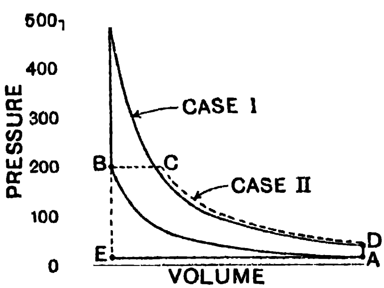


FIG. 623.

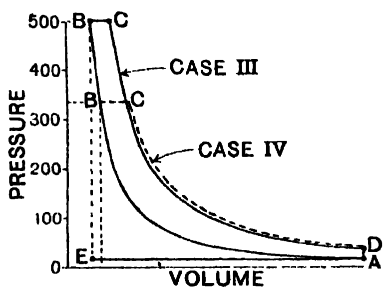


FIG. 624.

Heat received between B and C = $1172{\cdot}7K_v$.
Heat rejected between D and A = $(1033{\cdot}7 - 373)K_v = 660{\cdot}7K_v$.
Heat converted into work = $(1172{\cdot}7 - 660{\cdot}7)K_v = 512{\cdot}0K_v$.

Thermal efficiency $= \dfrac{512{\cdot}0K_v}{1172{\cdot}7K_v} = 0{\cdot}437$ or $43{\cdot}7$ per cent.

$$\text{M.E.P.} = \frac{P_B(V_C - V_B) + \dfrac{P_CV_C - P_DV_D}{n-1} - \dfrac{P_BV_B - P_AV_A}{n-1}}{V_A - V_B} = 61{\cdot}0$$

CASE III. *Diesel cycle* (Fig. 624).—$P_A = 15.$ $\quad P_B = P_C = 500.$

$P_BV_B^{1{\cdot}4} = P_AV_A^{1{\cdot}4}$, therefore, $V_B = V_A\left(\dfrac{P_A}{P_B}\right)^{\frac{1}{1{\cdot}4}} = 0{\cdot}0817V_A.$

$T_B = T_A\dfrac{P_BV_B}{P_AV_A} = 1015{\cdot}8$

The heat supplied between B and C is the same as in Cases I and II, namely $1172{\cdot}7K_v$.

Hence $(T_C - 1015{\cdot}8)K_p = 1172{\cdot}7K_v$,

therefore, $$T_C = \frac{1172{\cdot}7}{1{\cdot}4} + 1015{\cdot}8 = 1853{\cdot}4.$$

$V_C = V_B\dfrac{T_C}{T_B} = 0{\cdot}1491V_A$. $P_DV_D^{1\cdot4} = P_CV_C^{1\cdot4}$, hence $P_D = 34{\cdot}8$.

$\dfrac{T_D}{T_A} = \dfrac{P_D}{P_A}$, hence $T_D = 865{\cdot}4$.

Heat received between B and C $= 1172{\cdot}7K_v$.
Heat rejected between D and A $= (865{\cdot}4 - 373)K_v = 492{\cdot}4K_v$.
Heat converted into work $= (1172{\cdot}7 - 492{\cdot}4)K_v = 680{\cdot}3K_v$.

$$\text{Thermal efficiency} = \frac{680{\cdot}3K_v}{1172{\cdot}7K_v} = 0{\cdot}580 \text{ or } 58{\cdot}0 \text{ per cent.}$$

$$\text{M.E.P.} = \frac{P_B(V_C - V_B) + \dfrac{P_CV_C - P_DV_D}{n-1} - \dfrac{P_BV_B - P_AV_A}{n-1}}{V_A - V_B} = 74{\cdot}5.$$

CASE IV. *Diesel cycle.* (Fig. 624).—$P_A = 15$. $P_B = P_C = 335$.

Proceeding as in Case III the results are as follows.—

$V_B = 0{\cdot}1088V_A$. $T_B = 906{\cdot}3$ $T_C = 1744$.
$V_C = 0{\cdot}2094V_A$. $P_D = 37{\cdot}5$ $T_D = 932{\cdot}5$.
Thermal efficiency $= 0{\cdot}523$ or $52{\cdot}3$ per cent. M.E.P. $= 69{\cdot}2$.

The data and results of these four cases may now be tabulated. V_A, which is the same in each case, will be taken as 1 cubic foot.

Comparison of Ideal Otto and Diesel Cycles

Case.	Otto I	Diesel II.	Diesel III.	Diesel IV.
Maximum pressure P_C	500	200	500	335
Compression pressure P_B	200	200	500	335
Compression ratio $V_A \div V_B$	6·36	6·36	12·24	9·19
Expansion ratio $V_D \div V_C$	6·36	3·07	6·71	4·78
Maximum temperature $T_C - 273$. °C	1682	1346	1580	1471
Compression temperature $T_B - 273$. °C	509	509	743	633
Stroke volume $V_A - V_B$. c. ft.	0·843	0·843	0·918	0·891
Thermal efficiency . per cent.	**52·3**	**43·7**	**58·0**	**52·3**
M.E.P. $= P_m$ lb. per sq. in.	73·1	61·0	74·5	69·2
Work per cycle $= 144\ P_m(V_A - V_B)$ ft.-lb.	8874	7405	9848	8879
Stroke volume $(V_A - V_B)$ for the work 8874 ft.-lb. per cycle in each case	0·843	1·010	0·827	0·890
Relative fuel consumption for the same work per cycle	1·00	1·20	0·90	1·00

Comparing Cases I and II, in which the compression ratio is the same, it is seen that the Otto cycle is considerably more efficient than the Diesel cycle. In Case IV the efficiency of the Diesel cycle has been raised to equal that of the Otto cycle, by increasing the compression ratio from 6·36 to 9·19. In Case III the compression ratio has been raised to 12·24 causing the efficiency to rise to 58 per cent. In each case the total volume of the cylinder is the same, but the stroke volume is not the same, as is shown by the values of $V_A - V_B$. The second line from the bottom of the table shows what the stroke volume would have to be to make the work per cycle the same as in Case I and the bottom line shows, for this altered stroke volume, the relative fuel consumption which before was the same in each case.

The foregoing calculated results are of course ideal and do not agree with the results of actual practice. The temperatures T_B and T_C for instance are much higher than the corresponding temperatures in practice under similar conditions as to pressure and compression ratio. A comparison of the results in the different cases is nevertheless instructive.

376. Specific Heat of Gases in Foot-Pounds per Cubic Foot.—In dealing with solids and liquids it is most convenient to express specific heat as the amount of heat required to raise a pound weight of the substance one degree in temperature, the result being in C.H.U. or B.Th.U. according as the temperature scale is Centigrade or Fahrenheit. In dealing with gases, however, it is generally more convenient, in internal combustion engine problems, to express the specific heat in foot-pounds per cubic foot of gas at standard pressure (14·7 lb. per sq. in., or 760 mm., or 29·92 inches of mercury) and temperature (0° C.).

Conversion of specific heat from one system of units to the other is quite simple. Take the case of nitrogen for example. In C.H.U. the specific heat of nitrogen at constant pressure is, $K_p = 0{\cdot}235 + 0{\cdot}000{,}04t$, where t is the temperature in degrees centigrade. At standard pressure and temperature the weight of a cubic foot of nitrogen is 0·0781 lb. Hence the specific heat of one cubic foot of nitrogen is $(0{\cdot}235 + 0{\cdot}000{,}04t) \times 0{\cdot}0781$ C.H.U. But one C.H.U. is equal to 1400 ft.-lb., therefore,

$$K_p = (0{\cdot}235 + 0{\cdot}000{,}04t) \times 0{\cdot}0781 \times 1400 = 25{\cdot}69 + 0{\cdot}00437t \text{ ft.-lb.},$$

and the mean specific heat from 0° to $t°$ is, (see Art. 9, p. 6) $k_p = 25{\cdot}69 + 0{\cdot}00219t$ ft.-lb.

Specific heat at constant volume (K_v) is converted in the same way, that is, by multiplying by the weight of a cubic foot of the gas at standard pressure and temperature, and by 1400, the mechanical equivalent of heat.

In this new way of expressing the specific heat of a gas, the practice is to use the Centigrade scale of temperature exclusively.

For convenience of reference the specific heat of various gases, at constant volume, required in internal combustion engine problems are tabulated below in both systems of units.

Specific Heat of Gases at Constant Volume.

Gas.	C.H.U. (per lb.)	Ft.-lb per cubic foot.
N_2	$K_v = 0.165 + 0.000,04t.$ $k_v = 0.165 + 0.000,02t.$	$K_v = 18.04 + 0.00437t.$ $k_v = 18.04 + 0.00219t.$
O_2	$K_v = 0.149 + 0.000,04t.$ $k_v = 0.149 + 0.000,02t.$	$K_v = 18.61 + 0.0050t.$ $k_v = 18.61 + 0.0025t.$
Air	$K_v = 0.162 + 0.000,04t.$ $k_v = 0.162 + 0.000,02t.$	$K_v = 18.30 + 0.00452t.$ $k_v = 18.30 + 0.00226t.$
CO_2	$K_v = 0.150 + 0.000,117t.$ $k_v = 0.150 + 0.000,058t.$	$K_v = 26.0 + 0.0202t.$ $k_v = 26.0 + 0.0101t.$
H_2O	$K_v = 0.321 + 0.000,234t.$ $k_v = 0.321 + 0.000,117t.$	$K_v = 22.9 + 0.0167t$ $k_v = 22.9 + 0.0084t.$

It will be noticed that the specific heats of N_2, O_2, and air, expressed in foot-pounds per cubic foot, are nearly the same. Langen's formulæ for these gases and for CO, in foot-pounds per cubic foot, are

$$K_v = 18{\cdot}6 + 0{\cdot}00466t, \text{ and } k_v = 18{\cdot}6 + 0{\cdot}00233t.$$

The formulæ for CO_2 and H_2O in the above table are also Langen's.

In dealing with water vapour or steam (H_2O) it is usual, in internal combustion engine problems, to assume that at temperatures below the boiling point the H_2O remains in the gaseous state and behaves like the other gases. Hence the weight of a cubic foot of water vapour at standard pressure and temperature would be taken as $\frac{273 + 100}{273} \times 0{\cdot}0373 = 0{\cdot}051$ lb., where 0·0373 is the weight of a cubic foot of steam at 14·7 lb. per sq. in. and 100° C.

The formula for the specific heat of a mixture of gases may be found from the formulæ for the specific heats of the constituents as shown in the following example.

Example.—The products of combustion in a gas engine cylinder consist of N_2 and O_2, 0·83; CO_2, 0·05; and H_2O, 0·12, by volume, reduced to standard pressure and temperature. This may be taken as a normal gas engine mixture. Using Langen's formulæ for the constituents the formulæ for the above mixture are—

$$K_v = 0{\cdot}83(18{\cdot}6 + 0{\cdot}00466t) + 0{\cdot}05(26{\cdot}0 + 0{\cdot}0202t) + 0{\cdot}12(22{\cdot}9 + 0{\cdot}0167t) = 19{\cdot}49 + 0{\cdot}00688t, \text{ ft.-lb.}$$

and $k_v = 19{\cdot}49 + 0{\cdot}00344t$, ft.-lb.

The density of this mixture is 0·0783 lb. per cubic foot at standard

pressure and temperature. Hence, dividing the above expressions by $0{\cdot}0783 \times 1400$ the following formulæ are obtained,

$$K_v = 0{\cdot}178 + 0{\cdot}000{,}0628t, \text{ C.H.U.}$$
$$\text{and } k_v = 0{\cdot}178 + 0{\cdot}000{,}0314t, \text{ C.H.U.}$$

The value of R in the gas equation PV = RT is 99·03 for the above mixture, P being in lb. per sq. ft., V in cubic feet, and T absolute temperature Centigrade.

377. Specific Heat Formulæ for Normal Gas Engine Mixture.—The expressions for the specific heat at constant pressure and the specific heat at constant volume are, in general, sufficiently accurate when stated in the form,

$$K_p = a + st, \quad \text{and} \quad K_v = b + st. \qquad \text{But, } K_p - K_v = \frac{R}{J}$$

Hence, for the normal gas engine mixture, $a - b = \dfrac{99{\cdot}03}{1400}$ $= 0{\cdot}0707$ C.H.U.

Expressions for K_p and k_p corresponding to those for K_v and k_v determined in the latter part of the preceding Art., for the normal gas engine mixture, may now be found. Both sets of formulæ are here tabulated.

Specific Heats of Normal Gas Engine Mixture (Langen Formulæ).

C.H.U. (per lb.).	Ft.-lb. per cubic foot.
$K_p = 0{\cdot}248 + 0{\cdot}000{,}0628t.$ $k_p = 0{\cdot}248 + 0{\cdot}000{,}0314t.$	$K_p = 27{\cdot}24 + 0{\cdot}00688t.$ $k_p = 27{\cdot}24 + 0{\cdot}00344t.$
$K_v = 0{\cdot}178 + 0{\cdot}000{,}0628t.$ $k_v = 0{\cdot}178 + 0{\cdot}000{,}0314t.$	$K_v = 19{\cdot}49 + 0{\cdot}00688t.$ $k_v = 19{\cdot}49 + 0{\cdot}00344t.$

In the 1908 report of the Gaseous Explosions Committee of the British Association a curve is given showing the relation between the internal energy and the temperature for the normal gas engine mixture. From this curve Mr. H. E. Wimperis deduced the formula, $K_v = 0{\cdot}172 + 0{\cdot}000{,}075t$. This formula and those derived from it are here tabulated.

Specific Heats of Normal Gas Engine Mixture (Wimperis Formulæ).

C.H.U. (per lb.).	Ft.-lb. per cubic foot.
$K_p = 0{\cdot}243 + 0{\cdot}000{,}075t.$ $k_p = 0{\cdot}243 + 0{\cdot}000{,}0375t.$	$K_p = 26{\cdot}60 + 0{\cdot}0082t.$ $k_p = 26{\cdot}60 + 0{\cdot}0041t.$
$K_v = 0{\cdot}172 + 0{\cdot}000{,}075t.$ $k_v = 0{\cdot}172 + 0{\cdot}000{,}0375t.$	$K_v = 18{\cdot}85 + 0{\cdot}0082t.$ $k_v = 18{\cdot}85 + 0{\cdot}0041t.$

More recent formulæ for the specific heats of the normal gas engine mixture are those of Stodola given below.

$$\begin{aligned} K_p &= 0 \cdot 2541 + 0 \cdot 000{,}058t - 0 \cdot 000{,}000{,}00725t^2 \\ k_p &= 0 \cdot 2541 + 0 \cdot 000{,}029t - 0 \cdot 000{,}000{,}00242t^2 \\ K_v &= 0 \cdot 1834 + 0 \cdot 000{,}058t - 0 \cdot 000{,}000{,}00725t^2 \\ k_v &= 0 \cdot 1834 + 0 \cdot 000{,}029t - 0 \cdot 000{,}000{,}00242t^2 \end{aligned} \quad \text{C.H.U. (per lb.).}$$

$$\begin{aligned} K_p &= 27 \cdot 85 + 0 \cdot 00636t - 0 \cdot 000{,}000{,}795t^2 \\ k_p &= 27 \cdot 85 + 0 \cdot 00318t - 0 \cdot 000{,}000{,}265t^2 \\ K_v &= 20 \cdot 104 + 0 \cdot 00636t - 0 \cdot 000{,}000{,}795t^2 \\ k_v &= 20 \cdot 104 + 0 \cdot 00318t - 0 \cdot 000{,}000{,}265t^2 \end{aligned} \quad \text{Foot-pounds per cubic foot.}$$

378. Internal Energy of Normal Gas Engine Mixture.—The general formula for the mean specific heat of a gas from 0° C. to t° C. may be taken as $k_v = b + st$, ft.-lb., and the internal energy of the gas, above that which it contains at 0° C., is $(b + st)t$, ft.-lb. Taking the normal gas engine mixture, and the Langen formula for its k_v, the expression for the internal energy is $(19{\cdot}49 + 0{\cdot}00344t)t$, ft.-lb. per cubic foot. This is represented by the curve L in Fig. 625. The upper part of the British Association curve for the same mixture is shown at W and the upper part of the Stodola curve is shown at S. The curve S crosses the curve L at T and the curve W crosses the curve L at N. The lower parts of the curves W and S lie very near to that of L and for the sake of clearness they have been left out.

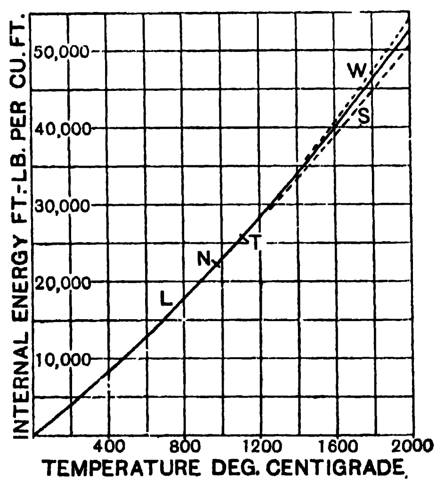


FIG. 625.—Internal energy curve for normal gas engine mixture

To be of use in solving problems the internal energy curve should be drawn to a large scale on squared paper.

EXAMPLE I.—Let it be required to find by how much the internal energy of a cubic foot of normal gas engine mixture is increased when its temperature is raised from 250° C. to 1400° C. at constant volume.

From the curve of Fig. 625 on a large scale the internal energy is 5100 ft.-lb. at 250° C. and 34,000 ft.-lb. at 1400° C. The increase in internal energy is therefore 34,000 − 5100 = 28,900 ft.-lb.

Or, by calculation :

Internal energy at 250° C. = $(19{\cdot}49 + 0{\cdot}00344 \times 250) \times 250 = 5087$

" " 1400° C. = $(19{\cdot}49 + 0{\cdot}00344 \times 1400) \times 1400 = 34{,}028$.

Therefore, increase in internal energy = 34,028 − 5087 = 28,941 ft.-lb.

EXAMPLE II.—A cubic foot of normal gas engine mixture at 240° C. has its internal energy increased by 30,100 ft.-lb. The volume remaining constant, it is required to find its new temperature.

From the chart the internal energy at 240° C. is 4900. Adding 30,100 to this gives 35,000 ft.-lb. as the energy at the new temperature, and from the chart the new temperature is found to be 1430° C.

Or, by calculation :

Internal energy at 240° C. $= (19{\cdot}49 + 0{\cdot}00344 \times 240) \times 240 = 4876$.

Internal energy at new temperature, t, $= 4876 + 30{,}100 = 34{,}976$.

Hence, $19{\cdot}49t + 0{\cdot}00344t^2 = 34{,}976$, a quadratic equation which, when solved, gives $t = 1432°$ C

For this type of problem the chart saves the labour of solving a quadratic equation.

379. Volumetric Efficiency.—If v denotes the volume, reduced to standard pressure and temperature, of air and gas, or air and fuel vapour, drawn in to the cylinder during a suction stroke, and if V denotes the volume swept through by the piston in one stroke, then v/V is the *volumetric efficiency.*

The volume v is made up of two parts, v_1 the volume of the gas or fuel vapour, and v_2 the volume of the air. In the case of gas engines the gas generally passes through a meter and v_1 is therefore readily determined, but v_2 is not so easily found. The air consumption is of course very much greater than the gas consumption and the direct measurement of the volume of air used is not very frequently made. The amount of air used may be determined from analyses of samples of the gas and the exhaust products, but with hit-and-miss governing the samples of the exhaust products should be taken during working cycles only if great accuracy is aimed at, and this requires the provision of a special mechanism to automatically close the sampling tube during idle cycles.

In four-stroke cycle gas engines working under normal conditions the volumetric efficiency is about 81 per cent. at full load. With hit-and-miss governing the volumetric efficiency increases as the load decreases, being about 84 per cent. at half-load, and about 88 per cent. at quarter-load.

380. Suction Temperature.—Although the exhaust and suction strokes may not take place at atmospheric pressure, the final pressure in these strokes is generally nearly atmospheric and in what follows it will be assumed that the final pressure in the exhaust and suction strokes is atmospheric.

A fresh charge entering the cylinder is heated in two ways. It is heated by contact with the hot metal surfaces. It is also heated by mixing with the hot gases which remain in the clearance space from the previous cycle. Imagine that the heating of the fresh charge by the hot metal surfaces takes place before the heating by mixing with the hot gases. The heat imparted by the hot metal surfaces must then be sufficient to raise the fresh charge to an absolute temperature T and expand it from the volume v to the volume V, where v and V have the meanings given to them in the preceding Art. (see

Fig. 626). Hence $\frac{V}{v} = \frac{T}{273}$, and for a given volumetric efficiency T is fixed.[1]

Let V_c denote the clearance volume, T_e the absolute temperature of the gases left in the clearance space, and T_s the final absolute temperature of the hot gases and fresh charge. The hot gases in the clearance space, on mixing with the fresh charge will shrink to a volume $V_c' = \frac{T_s}{T_e} V_c$ and

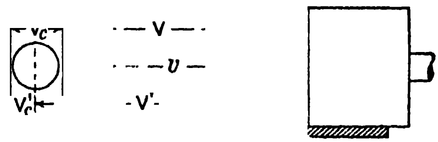


FIG. 626.

the fresh charge will expand to a volume $V' = \frac{T_s}{T} V$. But all the gases being at the same pressure,

$$V'_c + V' = \frac{T_s}{T_e} V_c + \frac{T_s}{T} V = V_c + V. \quad \text{Therefore, } T_s = \frac{V_c + V}{\frac{V_c}{T_e} + \frac{V}{T}}$$

and t_s (called the *suction temperature*) $= T_s - 273$.

It is usual to assume that $t_e = T_e - 273$ is the same as the temperature of the exhaust gases leaving the cylinder, a temperature which may readily be taken, but it should be taken as near to the cylinder as possible.

EXAMPLE.—Let $V_c = 0{\cdot}25V$, $t_e = 577°$ C., and $v/V = 0{\cdot}82$. Then,

$$T = 273\frac{V}{v} = \frac{273}{0{\cdot}82} = 333° \text{ C. absolute.}$$

$$T_s = \frac{V_c + V}{\frac{V_c}{T_e} + \frac{V}{T}} = \frac{0{\cdot}25V + V}{\frac{0{\cdot}25V}{850} + \frac{V}{333}} = 379° \text{ C. absolute} = 106° \text{ C.}$$

It is interesting to observe that for a given volumetric efficiency there is a certain minimum suction temperature. In the above example this minimum temperature is $333 - 273 = 60°$ C., and the amount by which the actual suction temperature exceeds the minimum depends on the temperature of the gases in the clearance space at the end of the exhaust stroke.

The results of the formula given above for the suction temperature have been plotted in Fig. 627 for three different volumetric efficiencies. These results are based on the assumption that $V_c = 0{\cdot}25V$, that is,

[1] If the absolute temperature of the charge, before entering the cylinder, is T′, its volume v' will be such that $\frac{v'}{v} = \frac{T'}{273}$. On receiving heat from the hot metal surfaces the volume v' will change to V and $\frac{V}{v'} = \frac{T}{T'}$. Hence $\frac{v'}{v} \times \frac{V}{v'} = \frac{T'}{273} \times \frac{T}{T'}$. Therefore, $\frac{V}{v} = \frac{T}{273}$ and the temperature T′ need not be considered.

that the compression ratio is 5 to 1. It is instructive to notice that, with a given volumetric efficiency, the variation in the suction temperature is very much smaller than the corresponding variation in the exhaust temperature. For instance, with a volumetric efficiency of 80 per cent. the suction temperature only rises 10° C. while the exhaust temperature rises from 400° C. to 600° C.

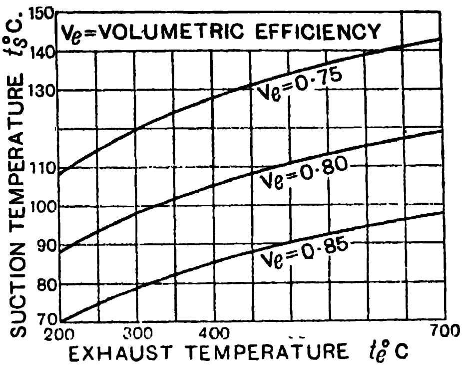


FIG. 627.

381. Adiabatic Expansion or Compression with Variable Specific Heats.—In what follows quantities of heat will be in foot-pounds per standard cubic foot of the gas or gaseous mixture and there will therefore be no need to introduce the mechanical equivalent of heat J.

The formulæ for the specific heats will be taken as

$$K_p = \alpha + sT, \text{ and } K_v = \beta + sT,$$

where $\alpha = a - 273s$, and $\beta = b - 273s$, a, b, and s being the constants in the formulæ $K_p = a + st$, and $K_v = b + st$ previously used.

Let a small quantity of heat δQ be given to (or taken from) a gaseous mixture whose pressure is P lb. per square foot, volume V cubic feet, and absolute temperature T. δP = change in pressure, δV = change in volume, and δT = change in temperature.

The change in internal energy is $K_v \delta T$, and the external work done is $P\delta V$.

$$\text{Hence, } \delta Q = K_v \delta T + P\delta V, \text{ and, } \frac{dQ}{dV} = K_v \frac{dT}{dV} + P.$$

$$\text{But, } \quad T = \frac{PV}{R}, \text{ therefore, } \frac{dT}{dV} = \frac{1}{R}\left(P + V\frac{dP}{dV}\right)$$

$$\text{Hence, } \frac{dQ}{dV} = \frac{K_v}{R}\left(P + V\frac{dP}{dV}\right) + P = \frac{\beta + sT}{\alpha - \beta}\left(P + V\frac{dP}{dV}\right) + P$$

$$= \frac{1}{\alpha - \beta}\left\{P(\alpha + sT) + V\frac{dP}{dV}(\beta + sT)\right\}$$

For adiabatic expansion or compression $\delta Q = 0$.

$$\text{Then, } P(\alpha + sT) + V\frac{dP}{dV}(\beta + sT) = 0.$$

Dividing by PV and inserting $\frac{PV}{R}$ for T,

$$\frac{1}{V}\left(a + s\frac{PV}{R}\right) + \frac{1}{P}\frac{dP}{dV}\left(\beta + s\frac{PV}{R}\right) = 0.$$

$$a\frac{dV}{V} + \frac{sP}{R}dV + \beta\frac{dP}{P} + \frac{sV}{R}dP = 0.$$

$$a\frac{dV}{V} + \beta\frac{dP}{P} + \frac{s}{R}(PdV + VdP) = 0.$$

$$a\frac{dV}{V} + \beta\frac{dP}{P} + \frac{s}{R}d(PV) = 0$$

Hence by integration,

$$a \log_e V + \beta \log_e P + \frac{s}{R} PV = C, \text{ a constant}$$

or, $a \log_e V + \beta \log_e P + sT = C.$

To use this for constructing the curve, first put

$$\log_e P = \log_e R + \log_e T - \log_e V, \text{ since } PV = RT.$$

Then $(a - \beta) \log_e V + \beta \log_e T + sT = C_1$

and $$\log_e V = \frac{C_1 - (\beta \log_e T + sT)}{a - \beta}$$

or for common logs., $\log V = \dfrac{C_1 - (2{\cdot}3026\beta \log T + sT)}{2{\cdot}3026(a - \beta)}$

In a similar manner, $\log P = \dfrac{2{\cdot}3026a \log T + sT - C_2}{2{\cdot}3026(a - \beta)}$

The values of the constants C, C_1, and C_2 are determined from a point on the curve for which P, V, and T are known.

For selected values of T the corresponding values of P and V may be calculated by means of the last two equations above.

Pressures in lb. per square inch (p) may be used instead of P as this simply involves a difference in the constant C or C_2, then for symmetry v may be used instead of V.

EXAMPLE.—A standard cubic foot of normal gas engine mixture, having an absolute temperature 2250° C., a pressure of 430 lb. per square inch, and a volume v_1, expands adiabatically until its volume is $5v_1$. It is required to construct the pressure-volume curve.

For this mixture, $a = 25{\cdot}36$, $\beta = 17{\cdot}61$, $a - \beta = 7{\cdot}75$, and $s = 0{\cdot}00688$.

Using common logs. and expressing volumes as multiples of v_1—

$$\begin{aligned} &2{\cdot}3026(a - \beta) \log v + 2{\cdot}3026\beta \log T + sT \\ &\quad = 2{\cdot}3026 \times 7{\cdot}75 \log 1 + 2{\cdot}3026 \times 17{\cdot}61 \log 2250 + 0{\cdot}00688 \times 2250 \\ &\quad = 151{\cdot}408, \text{ which is the value of the constant } C_1 \end{aligned}$$

Hence, $\log v = \dfrac{151{\cdot}408 - 40{\cdot}549 \log T - 0{\cdot}00688T}{17{\cdot}845}$

Also, $2{\cdot}3026(\alpha - \beta) \log p - 2{\cdot}3026\alpha \log T - sT$

$$= 2{\cdot}3026 \times 7{\cdot}75 \log 430 - 2{\cdot}3026 \times 25{\cdot}36 \log 2250 - 0{\cdot}00688 \times 2250$$

$$= -164{\cdot}233 = -C_2$$

Hence, $\log p = \dfrac{58{\cdot}394 \log T + 0{\cdot}00688T - 164{\cdot}233}{17{\cdot}845}$

Starting with $T = 2200°$ and taking other values of T at intervals of 100°, corresponding values of v and p are determined by means of the above formulæ, and plotting these the "pv" curve shown in Fig. 628 is obtained. The temperature curve, $T - 273 = t$ is also shown on the same base. The student is recommended to work out this example and construct the curves to a fairly large scale on squared paper.

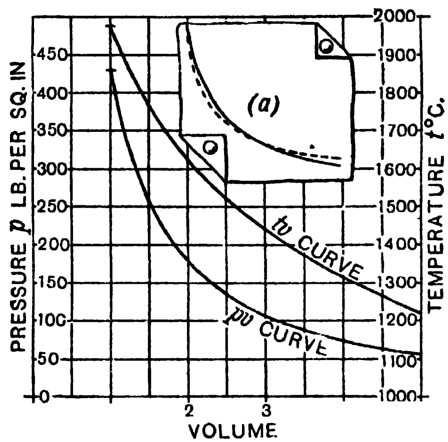


FIG. 628.

The exact temperature and pressure for the volume 5 can only be found by a series of trials, or by noting where the temperature and pressure curves cross the volume 5 line. In this example $T = 1490°$, very nearly, and $p = 56{\cdot}85$ lb. per square inch, for the volume 5.

382. The Curve $pv^n = C$ which approximates to the true Adiabatic.—Referring to the adiabatic of the preceding Article, a curve of the form $pv^n = C$ can be found which approximates to it and has under it the same area as that under the adiabatic.

If T_1 and T_2 are the absolute temperatures at the beginning and end of the adiabatic expansion, then the value of the index n in the equation $pv^n = C$ for the curve which approximates to the adiabatic is given very approximately by the formula $n = \dfrac{\alpha + \frac{s}{2}(T_1 + T_2)}{\beta + \frac{s}{2}(T_1 + T_2)}$.

Applying this formula to the example of the preceding Article,

$$n = \frac{25{\cdot}36 + 0{\cdot}00344(2250 + 1490)}{17{\cdot}61 + 0{\cdot}00344(2250 + 1490)} \quad 1{\cdot}25$$

Constructing the curve $pv^{1{\cdot}25} = C$, the difference between it and the adiabatic is too small to be clearly shown in Fig. 628, but if the student will add this new curve to his drawing of the adiabatic, it will be found that the relation of the one curve to the other will be as

shown, greatly exaggerated, at (*a*) in Fig. 628. The dotted curve at (*a*) is the curve $pv^{1\cdot 25} = C$, and the other is the adiabatic. The upper part of the adiabatic lies above, while the lower part lies below the other.

It will be instructive to compute the areas under the two curves. Taking the adiabatic as applying to a standard cubic foot of normal gas engine mixture, which was assumed in constructing the curve, the area under the curve represents the loss of internal energy during expansion from the temperature $2250 - 273 = 1977°$ C. to the temperature $1490 - 273 = 1217°$ C. This loss of internal energy is, using the formula previously given,

$$(19{\cdot}49 + 0{\cdot}00344 \times 1977) \times 1977 \\ - (19{\cdot}49 + 0{\cdot}00344 \times 1217) \times 1217 \qquad 163 \text{ ft.-lb.}$$

To compute the area under the curve $pv^{1\cdot 25} = C$, in ft.-lb. per standard cubic foot of mixture, it is necessary to know the actual volume v_1 at the beginning of the expansion. The standard cubic foot is measured at 0° C. and 14·7 lb. per square inch. At the beginning of the expansion the temperature is 2250° C. absolute and the pressure is 430 lb. per square inch, therefore $v_1 = \frac{2250}{273} \times \frac{14{\cdot}7}{430} = 0{\cdot}2818$ cubic foot.

The pressure at the end of the expansion for the curve $pv^{1\cdot 25} = C$, is, $p_2 = p_1\left(\frac{v_1}{v_2}\right)^{1\cdot 25} = 430\left(\frac{1}{5}\right)^{1\cdot 25} = 57{\cdot}51$ lb. per square inch.

The area under the $pv^{1\cdot 25} = C$ curve $\left(\text{given by } \frac{P_1V_1 - P_2V_2}{n - 1}\right)$ is,

$$\frac{144(430 \times 1 - 57{\cdot}51 \times 5) \times 0{\cdot}2818}{1{\cdot}25 - 1} = 23{,}122 \text{ ft.-lb.}$$

which is in close agreement with the area, 23,163 ft.-lb., under the adiabatic.

383. Ideal Efficiency of the Otto Cycle with Actual Mixture and Variable Specific Heats.—A method of determining the ideal efficiency of the Otto cycle, using the actual gaseous mixture and allowing for the variation of the specific heats with changes of temperature, will here be considered with reference to a numerical example.

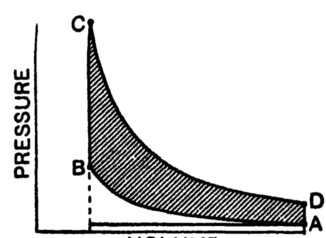


FIG. 629.

Consider a standard cubic foot of gaseous mixture containing 10 per cent., by volume, of coal gas having a calorific value of 300 C.H.U. per cubic foot, the remainder being air and products of combustion from the preceding cycle. This charge will be assumed to have a temperature of 100° C. and a pressure of 14·7 lb. per square inch at A, Fig. 629, the beginning of the compression stroke. The compression ratio will be taken as 5. For the compression curve

AB it is usual to assume that it follows the law $pv^{1\cdot37} = C$, which means that for this part of the cycle the variation in the specific heat is neglected. This is justified by the fact that before combustion the variation in the specific heat of the mixture is not so great as it is after combustion, also the increase in temperature is not so large. The index 1·37 corresponds to $K_v = 20\cdot95$ ft.-lb. per cubic foot.

With a compression ratio of 5 the pressure at B is $14\cdot7 \times 5^{1\cdot37} = 133\cdot3$ lb. per square inch, and the temperature is $373 \times 5^{0\cdot37} = 677°$ C. absolute or 404° C.

The internal energy at B is $404 \times 20\cdot95 = 8464$ ft.-lb.

The rise in temperature between A and B is $404 - 100 = 304°$ C., and the increase in internal energy between A and B is $304 \times 20\cdot95 = 6369$ ft.-lb. which is also the work of compression. The work of compression may also be found from the area under AB as follows. The volume of the standard cubic foot of mixture at B is

$$= \frac{677}{273} \times \frac{14\cdot7}{133\cdot3} = 0\cdot2735 \text{ cubic foot.}$$

Work of compression

$$\frac{144(133\cdot3 \times 1 - 14\cdot7 \times 5) \times 0\cdot2735}{1\cdot37 - 1} = 6365 \text{ ft.-lb.}$$

which agrees with the result obtained above in a much simpler way.

Combustion takes place at B and the amount of heat produced is $0\cdot1 \times 300 \times 1400 = 42{,}000$ ft.-lb.

The internal energy at C is $8464 + 42{,}000 = 50{,}464$ ft.-lb.

The usual chemical equations show that a standard cubic foot of mixture shrinks to less than a standard cubic foot of products after combustion. In this case it will be assumed that the volume of the products is 0·97 of a standard cubic foot. Hence, the internal energy of a standard cubic foot of products at C will be $\frac{50{,}464}{0\cdot97} = 52{,}025$ ft.-lb.

From the internal energy curve or by calculation as explained in Art. 378, p. 524, the temperature corresponding to this amount of internal energy is 1978° C. or say 2250° C. absolute.

The pressure at C is $133\cdot3 \times \frac{2250}{677} \times 0\cdot97 = 429\cdot7$, say 430 lb. per square inch. The multiplication by 0·97 allows for the shrinkage of volume mentioned above.

The next step is to find the temperature at the end of the adiabatic expansion CD. The construction of this curve has been fully explained in the preceding Article, but for the present purpose it is not necessary to construct it. All that need be done is to obtain a few points on the temperature-volume curve in the neighbourhood of the end of the expansion. The data of the example in the preceding Article are the same as the data for the adiabatic expansion curve in the present case and the temperature found at the end of the expansion was $1490 - 273 = 1217°$ C.

The internal energy per standard cubic foot at D is therefore,

$$(19\cdot49 + 0\cdot00344 \times 1217) \times 1217 = 28{,}814 \text{ ft.-lb.}$$

The area under the curve CD is equal to the difference between the internal energy at C and the internal energy at D, and if the work of compression be deducted from this difference the result is the net work done, shown by the hatched area ABCD in Fig. 629.

Therefore, net work done$=0{\cdot}97(52{,}025-28{,}814)-6369=16{,}146$ ft.-lb.

The heat expended for this amount of work has been found to be 42,000 ft.-lb.

The ideal efficiency is therefore, $\frac{16146}{42000}=0{\cdot}384$ or 38·4 per cent.

With a compression ratio of 5, as in this case, the air standard efficiency is 47·5 per cent.

384. Ideal Efficiency of Diesel Cycle with Actual Mixture and Variable Specific Heat.—The method used in this Article is on the lines of that used in the preceding one and it will be best understood by applying it to a numerical example. The calculations will be given in brief and the student should work out the missing steps and so verify the results given.

Referring to the ideal diagram of the cycle, Fig. 630, 1 lb. of air

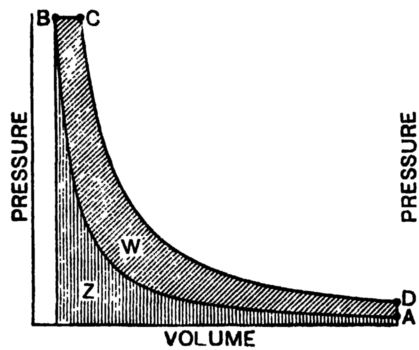


FIG. 630.

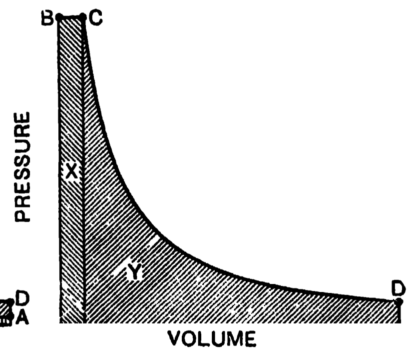


FIG. 631.

at A, the beginning of the compression stroke, is at a temperature of 350° C. absolute (77° C.) and has a pressure of 14·7 lb. per square inch. This air is compressed adiabatically as represented by AB, the compression ratio being 15. From B to C, at constant pressure, 0·03 lb. of fuel is introduced into the cylinder and burned. This fuel is accompanied by 0·05 lb. of additional air so that at C the mixture in the cylinder weighs 1·08 lb.

The composition of the fuel is: 86 per cent. of carbon, 12 per cent. of hydrogen, and 2 per cent. of incombustible matter which will be reckoned as nitrogen in the heat calculations. The lower calorific value of the fuel is 10,500 C.H.U. per lb.

From C to D the mixture expands adiabatically.

In considering the compression of the air the following specific heat formulæ will be used.

$$K_p=0{\cdot}230+0{\cdot}000{,}04t=0{\cdot}219+0{\cdot}000{,}04T.$$
$$K_v=0{\cdot}162+0{\cdot}000{,}04t=0{\cdot}151+0{\cdot}000{,}04T.$$

Hence, $a=0{\cdot}219$, $\beta=0{\cdot}151$, $a-\beta=0{\cdot}068$, and $s=0{\cdot}000{,}04$.

The equation to the adiabatic compression curve AB is

$$\alpha \log_e v + \beta \log_e p + s\mathrm{T} = \text{constant}.$$

From this equation, using common logs, and taking $v = 15$ at A, $v = 1$ at B, $\mathrm{T} = 350$ at A, and $p = 14{\cdot}7$ at A, the following formulæ are obtained:—

$$\log v = \frac{10{,}827{\cdot}48 - 3477 \log \mathrm{T} - 0{\cdot}4\mathrm{T}}{1566}$$

$$\log p = \frac{5043 \log \mathrm{T} + 0{\cdot}4\mathrm{T} - 11{,}141{\cdot}72}{1566}$$

After a few trials with the first of these two formulæ, T at B is found to be 998° C. absolute and therefore $t = 725°$ C.

Then by means of the second formula, p is found to be 628 lb. per square inch.

For the curve of the form $pv^n = \mathrm{C}$, which closely follows the true adiabatic compression curve, the index n is 1·38, obtained by means of the formula given in Art. 382, p. 529.

From the composition of the fuel and that of the air in which it is burned, the composition of the resulting mixture is found.

The following formulæ will be taken for the specific heats of the constituents of the mixture after combustion,

CO_2	$K_p = 0{\cdot}195 + 0{\cdot}000{,}117t.$	$K_v = 0{\cdot}150 + 0{\cdot}000{,}117t.$
H_2O	$K_p = 0{\cdot}430 + 0{\cdot}000{,}234t.$	$K_v = 0{\cdot}321 + 0{\cdot}000{,}234t.$
O_2	$K_p = 0{\cdot}212 + 0{\cdot}000{,}038t.$	$K_v = 0{\cdot}149 + 0{\cdot}000{,}038t.$
N_2	$K_p = 0{\cdot}235 + 0{\cdot}000{,}038t$	$K_v = 0{\cdot}165 + 0{\cdot}000{,}038t.$

By means of the above formulæ and the composition of the mixture after combustion the following formulæ for the specific heats of the mixture are found.

$$K_p = 0{\cdot}234 + 0{\cdot}000{,}050{,}8t = 0{\cdot}220 + 0{\cdot}000{,}050{,}8\mathrm{T}.$$
$$K_v = 0{\cdot}166 + 0{\cdot}000{,}050{,}8t = 0{\cdot}152 + 0{\cdot}000{,}050{,}8\mathrm{T}.$$

Hence, $\alpha = 0{\cdot}220$, $\beta = 0{\cdot}152$, $\alpha - \beta = 0{\cdot}068$, and $s = 0{\cdot}000{,}050{,}8$.

Just before the fuel and spraying air are introduced the specific heat at B is $K_p = 0{\cdot}230 + 0{\cdot}000{,}04 \times 725 = 0{\cdot}259$.

The specific heat at C is $K_p = 0{\cdot}234 + 0{\cdot}000{,}050{,}8t$, where t is the temperature at C.

The mean specific heat between B and C is therefore

$$0{\cdot}2465 + 0{\cdot}000{,}025{,}4t.$$

The heat in the 1 lb. of air at B is

$$(0{\cdot}162 + 0{\cdot}000{,}02 \times 725) \times 725 = 128 \text{ C.H.U.}$$

Assuming that the fuel and the spraying air enter at a temperature of 25° C., and taking the mean specific heat of the fuel as 0·6 and the mean specific heat of the spraying air as 0·24, the heat taken up by the fuel and spraying air on entering at B is

$$(725 - 25)(0{\cdot}03 \times 0{\cdot}6 + 0{\cdot}05 \times 0{\cdot}24) = 21 \text{ C.H.U.}$$

The heat supplied by the combustion of the fuel between B and C is $10,500 \times 0 \cdot 03 = 315$ C.H.U

Hence,

$$128 + (0 \cdot 2465 + 0 \cdot 000{,}025{,}4t)(t - 725) \times 1 \cdot 08 = 315 + 128 - 21.$$

This is a quadratic equation which gives $t = 1667°$ C. Therefore $T_C = 1940°$ C.

Internal energy at C

$$= (0 \cdot 166 + 0 \cdot 000{,}025{,}4 \times 1667) \times 1667 \times 1 \cdot 08 = 375 \text{ C.H.U.}$$

External work done between B and $C = 128 + 315 - 375 = 68$ C.H.U.

Volume of 1 lb. of air at $A = 12 \cdot 39 \times \frac{350}{273} = 15 \cdot 885$ cubic feet.

$$\text{Volume of 1 lb. of air at B} = \frac{15 \cdot 885}{15} = 1 \cdot 059 \text{ cubic feet.}$$

Let v = volume of 1·08 lb. of mixture at C. Then external work done between B and $C = 628 \times 144(v - 1 \cdot 059)$ ft.-lb.

Therefore $628 \times 144(v - 1 \cdot 059) = 68 \times 1400$. Hence, $v = 2 \cdot 112$ c. ft.

The equation to the adiabatic expansion curve CD is,

$$a \log_e v + \beta \log_e p + sT = \text{constant.}$$

From this equation, using common logs, and taking $v = 2 \cdot 112$ at C, $v = 15 \cdot 885$ at A, $T = 1940$ at C, and $p = 628$ at C, and the values of a, β, and s already found, the following formulæ are obtained:

$$\log v = \frac{13001 \cdot 3 - 3500 \log T - 0 \cdot 508T}{1566}$$

$$\log p = \frac{5066 \log T + 0 \cdot 508T - 13259 \cdot 92}{1566}$$

After a few trials with the first of these two formulæ, T at D is found to be 1057° C. absolute and therefore $t = 784°$ C.

Then by means of the second formula, p at D is found to be 45·5 lb. per square inch.

For the curve of the form $pv^n = C$, which closely follows the true adiabatic expansion curve, the index n is 1·30, obtained by means of the formula given in Art. 382.

Internal energy at $D = (0 \cdot 166 + 0 \cdot 000{,}025{,}4 \times 784) \times 784 \times 1 \cdot 08 = 157 \cdot 4$ C.H.U.

Internal energy at C has been found to be 375 C.H.U.

Work of expansion $= 375 - 157 \cdot 4 = 217 \cdot 6$ C.H.U. = area Y, Fig. 631.

Work done between B and C = 68 C.H.U. = area X, Fig. 631.

Internal energy at $A = (0 \cdot 162 + 0 \cdot 000{,}02 \times 77) \times 77 = 12 \cdot 6$ C.H.U.

Internal energy at B = 128 C.H.U.

Work of compression $= 128 - 12 \cdot 6 = 115 \cdot 4$ C.H.U. = area Z, Fig. 630.

Net work done $= 68 + 217 \cdot 6 - 115 \cdot 4 = 170 \cdot 2$ C.H.U. = area W, Fig. 630.

Heat supplied = 315 C.H.U.

Ideal efficiency $= \frac{170 \cdot 2}{315} = 0 \cdot 54$ or 54 per cent.

Compression ratio = 15. Expansion ratio $= \frac{15 \cdot 885}{2 \cdot 112} = 7 \cdot 521$.

Using the formula found in Art. 374, the air standard efficiency is 60·4 per cent.

385. Entropy Formulæ with Variable Specific Heats.—The term *entropy* has been defined and explained in Art. 55, p. 59, but the formulæ there obtained were based on the assumption of constant specific heat. The subject will now be further considered and allowance made for the variation of the specific heats with change of temperature.

In what follows, to avoid confusion, quantity of heat will be expressed in heat units (say C.H.U.).

Let $K_p = \alpha + sT$, and $K_v = \beta + sT$.

For a small change in the heat of the substance

$$\delta Q = K_v \delta T + \frac{P \delta V}{J}, \text{ and } \delta \phi = \frac{\delta Q}{T} = K_v \frac{\delta T}{T} + \frac{P \delta V}{JT}.$$

But $PV = RT = (K_p - K_v) JT$, therefore $\frac{P}{JT} = \frac{K_p - K_v}{V}$.

Hence, $\delta \phi = K_v \frac{\delta T}{T} + (K_p - K_v) \frac{\delta V}{V}$

and in the limit $d\phi = K_v \frac{dT}{T} + (K_p - K_v) \frac{dV}{V}$

$$= \beta \frac{dT}{T} + s dT + (\alpha - \beta) \frac{dV}{V}$$

Integrating between the limits T_1 and T_2

$$\phi_2 - \phi_1 = \beta \log_e \frac{T_2}{T_1} + s(T_2 - T_1) + (\alpha - \beta) \log_e \frac{V_2}{V_1} \quad (1)$$

Since $\frac{P_2 V_2}{T_2} = \frac{P_1 V_1}{T_1}$, therefore, $\log_e \frac{V_2}{V_1} = \log_e \frac{T_2}{T_1} - \log_e \frac{P_2}{P_1}$

Hence, $\phi_2 - \phi_1 = \alpha \log_e \frac{T_2}{T_1} + s(T_2 - T_1) - (\alpha - \beta) \log_e \frac{P_2}{P_1}$ (2)

For a change of state at *constant volume* it follows from (1) that

$$\phi_2 - \phi_1 = \beta \log_e \frac{T_2}{T_1} + s(T_2 - T_1) \quad . \quad . \quad . \quad (3)$$

For a change of state at *constant pressure* it follows from (2) that

$$\phi_2 - \phi_1 = \alpha \log_e \frac{T_2}{T_1} + s(T_2 - T_1) \quad . \quad . \quad . \quad (4)$$

For a change of volume at *constant temperature* it follows from (1) that

$$\phi_2 - \phi_1 = (\alpha - \beta) \log_e \frac{V_2}{V_1} \quad . \quad . \quad . \quad . \quad (5)$$

For a change of pressure at *constant temperature* it follows from (2) that

$$\phi_2 - \phi_1 = -(\alpha - \beta) \log_e \frac{P_2}{P_1} \quad . \quad . \quad . \quad . \quad (6)$$

For *constant specific heat* $\alpha = K_p$, $\beta = K_v$, and $s = 0$, then the foregoing equations become—

$$\phi_2 - \phi_1 = K_v \log_e \frac{T_2}{T_1} + (K_p - K_v) \log_e \frac{V_2}{V_1} \quad . \quad . \quad (1')$$

$$\phi_2 - \phi_1 = K_p \log_e \frac{T_2}{T_1} - (K_p - K_v) \log_e \frac{P_2}{P_1} \quad . \quad . \quad (2')$$

At constant volume $\phi_2 - \phi_1 = K_v \log_e \frac{T_2}{T_1}$ . . . . (3')

At constant pressure $\phi_2 - \phi_1 = K_p \log_e \frac{T_2}{T_1}$ . . . . (4')

At constant temperature $\phi_2 - \phi_1 = (K_p - K_v) \log_e \frac{V_2}{V_1}$. (5')

$$= -(K_p - K_v) \log_e \frac{P_2}{P_1} \quad (6')$$

It will be observed that in all the foregoing formulæ Napierian logarithms are used. The Napierian log of a number may be obtained by multiplying the common log of the number by 2·3026.

386. Temperature-Entropy Diagram.—To construct the temperature-entropy or $T\phi$ diagram from the indicator or PV diagram the first step is to select a point on the latter diagram for which the temperature is known. The volume of 1 lb. of mixture in the cylinder for the point selected is then calculated from the known volume at standard pressure and temperature. This fixes the scale of volumes for the PV diagram. A number of points are next selected on the PV diagram, and for these the temperatures and entropies are calculated, and these when plotted give the contour of the $T\phi$ diagram

The calculation of the temperatures and entropies may be avoided by using a $T\phi$ chart, to be explained in the next Article, but it will be instructive to dispense with the chart for the present.

The case of an ideal PV diagram of the Otto cycle, with adiabatic compression and expansion, and with air as the working fluid, will first be considered. In this case the specific heats will be assumed to be constant.

The PV diagram is shown in Fig. 632. The point first selected is A at the beginning of the compression stroke where the pressure is 15 lb. per square inch and the temperature assumed to be 100° C. or

373° C. absolute. At standard pressure and temperature the volume of 1 lb. of air is 12·39 cubic feet. The volume of 1 lb. of air at A is therefore $12{\cdot}39 \times \frac{14{\cdot}7}{15} \times \frac{373}{273} = 16{\cdot}59$ cubic feet, which fixes the scale of volumes. In this case the compression ratio is 5 and it will be simpler to take the relative volumes instead of the actual volumes, the clearance volume being taken as 1.

PV = RT for all points on the PV diagram, therefore for any point x, $\frac{p_x v_x}{T_x} = \frac{p_A v_A}{T_A} = \frac{15 \times 5}{373}$, and $T_x = 4{\cdot}973 p_x v_x$.

The relative volumes and the pressures are tabulated for the selected points and also the values of T calculated from the foregoing equation. In this case no intermediate points are selected on the compression or expansion curves because, these being adiabatics, the

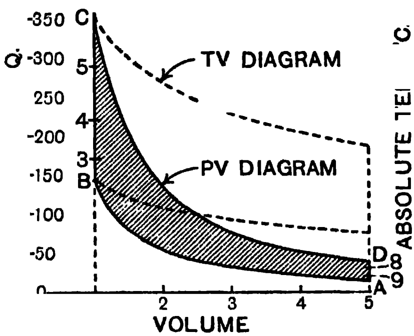


Fig. 632.

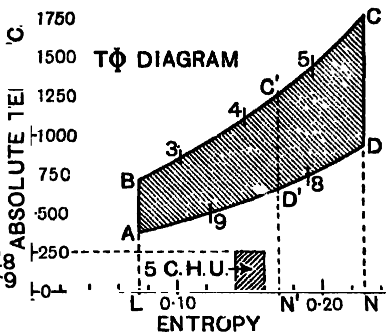


Fig. 633.

entropy for all points on AB is the same as at A, and the entropy for all points on CD is the same as at C.

Pt. x	A	B	3	4	5	C	D	8	9
Rel. V	5	1	1	1	1	1	5	5	5
p	15	142·8	170	220	290	357	37·5	30	20
T	373	710	845	1094	1442	1775	932	746	497
$\phi_x - \phi_A$	0·0	0·0	0·029	0·073	0·120	0·155	0·155	0·117	0·048
ϕ	0·073	0·073	0·102	0·146	0·193	0·228	0·228	0·190	0·121

The zero of entropy is taken at 0° C. or 273° C. absolute.

For air the specific heats are taken as, $K_p = 0{\cdot}237$, and $K_v = 0{\cdot}169$.

Denoting the entropy of air at 273° C. absolute, and 14·7 lb. per square inch, by ϕ_0. Then by equation (2′), p. 536, adapted to common logs.

$$\phi_A - \phi_0 = 2{\cdot}3026 K_p \log \frac{T_A}{273} - 2{\cdot}3026(K_p - K_v) \log \frac{p_A}{14{\cdot}7}$$

But $\phi_0 = 0$, therefore the entropy at A is

$$\phi_A = 0{\cdot}546 \log \frac{373}{273} - 0{\cdot}157 \log \frac{15}{14{\cdot}7} = 0{\cdot}073 = \phi_B$$

Changes along BC are changes at constant volume, therefore by equation (3′), p. 536, adapted to common logs—

$\phi_3 - \phi_B = 2{\cdot}3026 \times 0{\cdot}169 \log \frac{845}{710} = 0{\cdot}029 = \phi_3 - \phi_A$.
And $\phi_3 = 0{\cdot}073 + 0{\cdot}029 = 0{\cdot}102$.

In like manner the entropies at 4, 5, and C are found as tabulated, and $\phi_D = \phi_C$.

Changes along DA are changes at constant volume, therefore by equation (3′), adapted to common logs—

$\phi_8 - \phi_A = 2{\cdot}3026 \times 0{\cdot}169 \log \frac{746}{373} = 0{\cdot}117$.
And $\phi_8 = 0{\cdot}073 + 0{\cdot}117 = 0{\cdot}190$.
In like manner ϕ_9 is found.

The values of T and ϕ found are plotted, and the resulting $T\phi$ diagram is as shown in Fig. 633. The hatched area in Fig. 633 represents the work done during a cycle, in heat units (C.H.U.) per lb. of air in the cylinder, and the hatched area in Fig. 632 represents the same amount of work in foot-pounds. If both areas be expressed in terms of the same unit, say, ft.-lb., the results should be the same.

The area LBCN, Fig. 633, represents the heat supplied per lb. of air per cycle and the efficiency is $= \dfrac{\text{area ABCD}}{\text{area LBCN}}$.

In Art. 370, p. 515, it was shown that the air standard efficiency for the Otto cycle was $1 - \dfrac{T_D - T_A}{T_C - T_B} = 1 - \dfrac{T_D}{T_C} = 1 - \dfrac{T_A}{T_B}$. Hence, efficiency is $= \dfrac{T_C - T_D}{T_C} = \dfrac{CD}{CN}$ (Fig. 633) $= \dfrac{T_B - T_A}{T_B} = \dfrac{BA}{BL}$ (Fig. 633).

The student should now have no difficulty in proving that if any ordinate C′D′N′ be drawn in Fig. 633 the ratio of C′D′ to C′N′ is constant.

Another example will now be worked out on the lines of the preceding one, except that an actual indicator diagram will be used, and a normal gas engine mixture will be taken as the working fluid, also the specific heats will be assumed to vary with the temperature.

The indicator diagram is given in Fig. 634, and to enable the student to work out this example fully the volumes and pressures at a sufficient number of points are given so that the diagram may be accurately redrawn to enlarged scales and further selected points taken where thought necessary.

The point A at the beginning of the compression stroke is taken as the starting point. Here the pressure is 14·7 lb. per square inch and the absolute temperature 370° C.

The specific heats of the mixture are, $K_p = 0{\cdot}231 + 0{\cdot}000{,}0628\ T$, and $K_v = 0{\cdot}161 + 0{\cdot}000{,}0628T$. Therefore $a = 0{\cdot}231$, $\beta = 0{\cdot}161$, $a - \beta = 0{\cdot}07$, and $s = 0{\cdot}000{,}0628$.

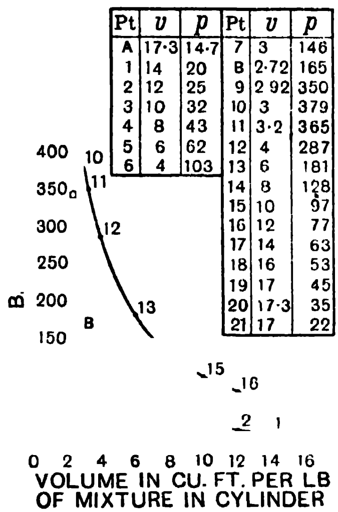
For adaptation of the entropy formulæ to common logarithms, $2{\cdot}3026\alpha = 0{\cdot}532$, $2{\cdot}3026\beta = 0{\cdot}371$, and $2{\cdot}3026(\alpha - \beta) = 0{\cdot}161$.

The volume of 1 lb. of mixture at standard pressure and temperature is 12·77 cubic feet.

Volume at A $= 12{\cdot}77 \times \frac{370}{273} = 17{\cdot}3$ cubic feet, and from this and the compression ratio 6·37 the volume scale may be constructed.

For all points on the PV diagram PV = RT, therefore for any point x, $\frac{p_x v_x}{T_x} = \frac{p_A v_A}{T_A} = \frac{14{\cdot}7 \times 17{\cdot}3}{370}$ and $T_x = 1{\cdot}455 p_x v_x$.

For example take the point 13. $T_{13} = 1{\cdot}455 \times 181 \times 6 = 1580$.



Pt	v	p	Pt	v	p
A	17·3	14·7	7	3	146
1	14	20	8	2·72	165
2	12	25	9	2·92	350
3	10	32	10	3	379
4	8	43	11	3·2	365
5	6	62	12	4	287
6	4	103	13	6	181
			14	8	128
			15	10	97
			16	12	77
			17	14	63
			18	16	53
			19	17	45
			20	17·3	35
			21	17	22

FIG. 634.

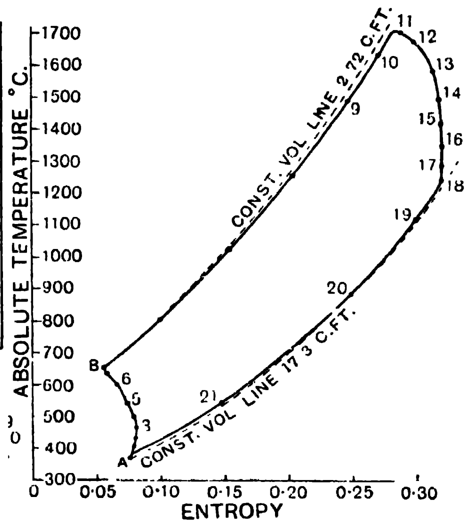


FIG. 635.

This calculation must be made for all the selected points and the results tabulated.

By equation (4), p. 535, adapted to common logs, the entropy at A is,

$$\phi_A = 0{\cdot}532 \log \tfrac{370}{273} + 0{\cdot}000{,}0628(370 - 273) = 0{\cdot}0763.$$

The value of $\phi_x - \phi_A$ for any other point x may be computed by means of equation (1) or equation (2), p. 535.

Taking equation (1), adapted to common logs, and applying it to point 13,

$$\phi_{13} - \phi_A = 0{\cdot}371 \log \frac{1580}{370} + 0{\cdot}000{,}0628(1580 - 370)$$
$$+ 0{\cdot}161 \log \frac{6}{17{\cdot}3} = 0{\cdot}2358,$$

and $\phi_{13} = 0{\cdot}2358 + 0{\cdot}0763 = 0{\cdot}3121$.

This calculation must be made for all the selected points and the results tabulated.

The form of the table and the results for six selected points are here given.

Point x	v	p	T	log v	log T	$\phi_x - \phi_A$	ϕ
A	17·3	14·7	370	1·23805	2·56820	0·0000	0·0763
3	10	32	466	1·00000	2·66839	0·0049	0·0812
9	2·92	350	1487	0·46538	3·17231	0·1699	0·2462
13	6	181	1580	0·77815	3·19866	0·2358	0·3121
16	12	77	1344	1·07918	3·12840	0·2434	0·3197
20	17·3	35	881	1·23805	2·94498	0·1719	0·2482

v is in cubic feet, p is in lb. per square inch, and T is in degrees Centigrade absolute.

The values of T and ϕ for the selected points are plotted in Fig. 635 and the Tϕ diagram obtained. To save space the absolute zero of temperature is outside the figure.

The constant volume lines for the volumes 2·72 cubic feet and 17·3 cubic feet have been added. As would be expected from an examination of Fig. 634, the explosion curve in Fig. 635 falls away from the constant volume line 2·72 as it rises.

Referring to Fig. 635 it will be seen that during compression the entropy increases from A to 3 and then diminishes from 3 to B. This shows that the mixture has been receiving heat from the cylinder between A and 3 and rejecting heat to the cylinder between 3 and B. Examining the expansion curve it will be seen that the entropy increases until the point 16 is reached. This shows that the mixture has been receiving heat up to this point, and since during this part of the cycle the mixture is probably hotter than the greater part of the cylinder, the heat received is most probably due to the combustion of the gas. Combustion is therefore probably not complete until the point 16 is reached. From 16 to 18, before the exhaust valve opens, the expansion is practically adiabatic.

387. Temperature-Entropy Chart for Normal Gas Engine Mixture.—It will have been seen that in the method of finding points on the Tϕ diagram from points on the PV diagram described in the preceding Art. the chief labour is in calculating the entropies for the different points. This labour may be avoided by using a chart which is made up of a number of constant volume and constant pressure lines. If for a particular point on the PV diagram the volume per lb. of mixture in the cylinder is, say, 3 cubic feet, and the pressure is, say, 150 lb. per square inch, the corresponding point on the Tϕ diagram will be on the chart at the intersection of the constant volume line 3 with the constant pressure line 150. It will be found that all the constant volume lines are of exactly the same shape so that a templet which fits one will fit all. This is also true of the constant pressure lines, although they differ from the constant volume lines.

The construction of such a chart for the normal gas engine mixture will now be described. The constants required in the entropy equations have been given in the preceding Art. and are here repeated, namely,

$$a = 0{\cdot}231,\ \beta = 0{\cdot}161,\ a - \beta = 0{\cdot}07,\ \text{and}\ s = 0{\cdot}000{,}0628$$

and for adaptation to common logs,

$$2{\cdot}3026\alpha = 0{\cdot}532, \quad 2{\cdot}3026\beta = 0{\cdot}371, \quad \text{and } 2{\cdot}3026(\alpha - \beta) = 0{\cdot}161.$$

The volume of the mixture at standard pressure and temperature is 12·77 cubic feet.

For the normal gas engine mixture equation (1), p. 535, adapted to common logs, is—

$$\phi_2 - \phi_1 = 0{\cdot}371 \log \frac{T_2}{T_1} + 0{\cdot}000{,}0628(T_2 - T_1) + 0{\cdot}161 \log \frac{v_2}{v_1}.$$

This mixture at 273° C. absolute, 12·77 cubic feet, and 14·7 lb. per square inch is taken as having zero entropy.

If $v_2 = v_1$ then $\phi_2 - \phi_1 = 0{\cdot}371 \log \frac{T_2}{T_1} + 0{\cdot}000{,}0628(T_2 - T_1)$ gives the difference between the entropies at any two temperatures T_2 and T_1 for the same constant volume line.

For the 12·77 cubic feet line, $\phi_1 = 0$ when $T_1 = 273$. Hence if $\phi_{12\cdot77}$ denotes the entropy for this line at temperature T then,

$$\begin{aligned}\phi_{12\cdot77} &= 0{\cdot}371 \log \frac{T}{273} + 0{\cdot}000{,}0628(T - 273)\\ &= 0{\cdot}371 \log T + 0{\cdot}000{,}0628T - 0{\cdot}9210.\end{aligned}$$

Let ϕ_{10} denote the entropy at temperature T for any point on the 10 cubic feet line, then by equation (5), p. 536, adapted to common logs—

$$\phi_{10} - \phi_{12\cdot77} = 0{\cdot}161 \log \frac{10}{12{\cdot}77} = -\ 0{\cdot}0171, \text{ at temperature T,}$$

Therefore,

$$\phi_{10} = \phi_{12\cdot77} - 0{\cdot}0171 = 0{\cdot}371 \log T + 0{\cdot}000{,}0628T - 0{\cdot}9381,$$

which is the equation to the 10 cubic feet constant volume line, and selecting values of T any number of points on this line can be found and the line may then be drawn. In Fig. 636 this line is marked v 10.

If ϕ_x be the entropy at temperature T for a point on the v_x constant volume line, and ϕ_y be the entropy at the same temperature T for a point on the v_y constant volume line then,

$$\phi_x - \phi_y = 0{\cdot}161 \log \frac{v_x}{v_y}.$$

If $\frac{v_x}{v_y} = 10$ or $\frac{1}{10}$, then $\phi_x - \phi_y = \pm\ 0{\cdot}161$, respectively.

Hence the constant volume lines, 0·1 cubic foot, 1 cubic foot, 10 cubic feet, and 100 cubic feet, are, at the same temperature, at intervals of 0·161 measured on the entropy scale. Hence after finding the 10 cubic feet line the others may be drawn by means of a templet made out of a piece of veneer or a sheet of celluloid, and used with a Tee-square, as shown in Fig. 637.

Intermediate curves are obtained by dividing the spaces between

the above mentioned curves logarithmically as indicated in Fig. 638, where the divisions along MN are taken from the divisions on a slide-rule and mN is the interval between two of the curves representing volumes which are in the ratio of 1 to 10.

In Fig. 636, for the sake of clearness, only the principal constant

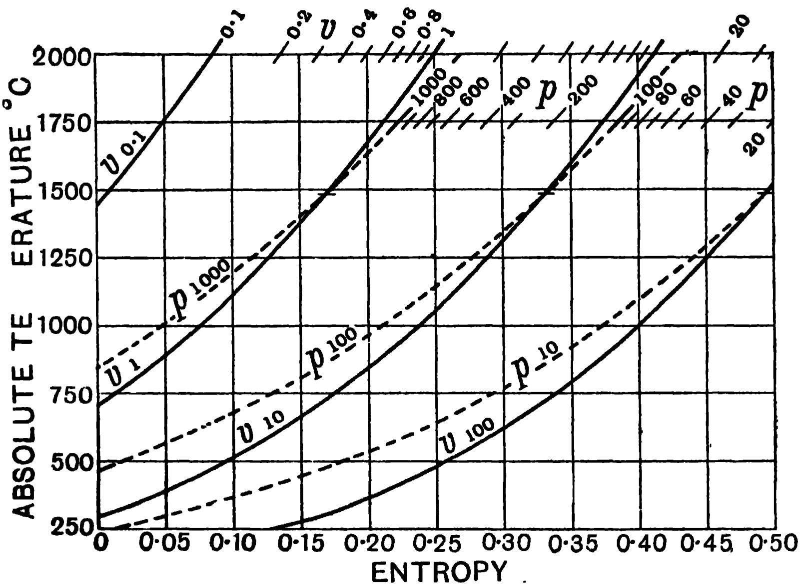


FIG. 636.—Temperature-entropy chart.

volume lines are shown, but on the actual chart numerous intermediate lines should be drawn and coloured inks should be used in inking them in.

The constant pressure lines are dealt with in exactly the same way as the constant volume lines.

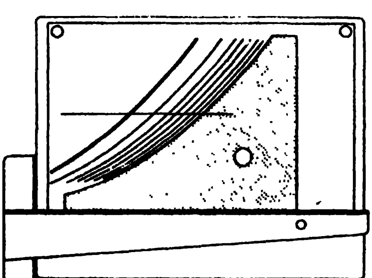


FIG. 637.

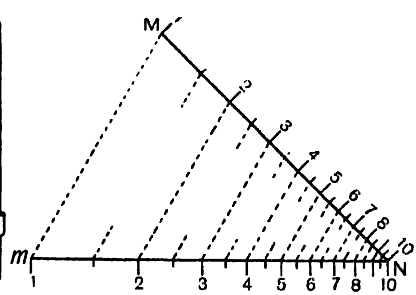


FIG. 638.

The equation to the constant pressure line 10 lb. per square inch is,

$$\phi_{10} = 0{\cdot}532 \log T + 0{\cdot}000{,}0628T - 1{\cdot}2862.$$

and the intervals between the constant pressure lines 10, 100, and 1000 lb. per square inch at the same temperature are each equal to 0·161 measured on the entropy scale. The intervals for intermediate lines are found in the same way as for the intermediate constant volume lines.

The chart must be constructed with great care on good squared paper and is permanent for the particular mixture for which it is drawn. Problems are worked on tracing paper placed over the chart.

Since the constant volume lines intersect the constant pressure lines at acute angles, the points of intersection are not very well defined, and a small error in the position of one of these intersecting lines causes a much larger error in the position of their point of intersection. This difficulty is overcome by making use of the temperature which is easily calculated.

It is frequently assumed that what has been called the normal gas engine mixture is the mixture for any internal combustion engine.

Exercises XXIII

1. The diameter of a gas engine cylinder is 11 inches, the stroke is 19 inches, and the clearance volume is 370 cubic inches. The temperature at the end of admission is 95° C. (203° F.), and the pressure is 15 lb. per sq. in. absolute. The compression curve conforms to the equation $pv^{1.37} =$ constant. Determine the pressure and temperature at the end of compression.

2. The cylinder of a four-stroke cycle gas engine has a clearance volume 1-4th of the volume swept through by the piston in one stroke. Assuming that the exhaust and suction pressures are 15 lb. per sq. in. absolute and that the compression and expansion curves follow the law $pv^{1.35} =$ constant, and that the maximum pressure is 2·5 times the pressure at the end of compression, calculate the maximum compression and explosion pressures and temperatures, the temperature at the beginning of the compression being 95° C. Also, assuming that the expansion curve is continued to the end of the stroke, calculate the pressure and temperature at that point. Lastly, compute the mean effective pressure for one cycle.

3. A gas engine working on the Otto cycle has a cylinder 8·5 inches in diameter × 18 inches stroke. The clearance volume is 0·2467 cubic foot. The weight of mixed gas and air per stroke is 0·0484 pound and its temperature at the end of the suction stroke is 62·8° C. at a pressure of 13·8 lb. per square inch absolute. The absolute pressures are :—At the end of compression, 67·8 lb. per square inch. At the end of constant volume explosion, 240 lb. per square inch. The volume at the end of constant pressure expansion is 0·2617 cubic foot.

Find: (1) The temperatures at the end of (*a*) compression, (*b*) explosion, (*c*) constant pressure expansion. (2) Heat added as shown by the diagram, (*a*) during explosion, (*b*) during constant pressure expansion. $K_p = 0{\cdot}26$. $K_v = 0{\cdot}19$. [U.L.]

4. A gas engine uses 16 cubic feet of gas per I.H.P. per hour. The lower calorific value of the gas is 270 C.H.U. per cubic foot. The compression ratio is 4·85. Calculate, (*a*) the indicated thermal efficiency, (*b*) the air standard efficiency, and (*c*) the relative efficiency.

5. What must be the compression ratio in a Diesel engine if the temperature at the end of the compression is 100° C. above the ignition temperature of the oil vapour if the latter is 515° C.? Assume that the temperature of the air at the beginning of the compression is 100° C., and its pressure 14·5 lb. per sq. in. absolute, and that the compression curve follows the law $pv^{1.33} =$ constant. Find also the pressure at the end of the compression.

6. In a Diesel engine the compression ratio is 12·3 and the expansion ratio

6·8. What is the air standard efficiency? If the actual indicated thermal efficiency is 41 per cent., what is the relative efficiency? If the calorific value of the oil used is 10,600, C.H.U., what is the consumption of oil in lb. per I.H.P. per hour?

7. A Diesel engine has a relative efficiency of 0·55 on the brake. If the compression ratio is 13·8 and the expansion ratio 7·4 and the lower calorific value of the oil is 10,500 C.H.U., find the consumption of oil in lb. per B.H.P. per hour. [U.L.]

8. Convert the following formulæ into formulæ in terms of T, absolute temperature Centigrade, t being ordinary temperature Centigrade. (1) $K_v = 19{\cdot}49 + 0{\cdot}00688t$. (2) $K_v = 0{\cdot}178 + 0{\cdot}000{,}0628t$. (3) $K_p = 0{\cdot}2541 + 0{\cdot}000{,}058t - 0{\cdot}000{,}000{,}00725t^2$.

9. If the heat required to raise 1 lb. of CO_2 from 1000° C. to 1001° C. is 0·2954 C.H.U. at constant pressure, what would the amount be at constant volume? Also express both quantities in foot-pounds per volume of gas, which at 0° C. and 14·7 lb. per square inch pressure is 1 cubic foot. Given that 1 cubic foot of CO_2 at 0° C. and 14·7 lb. per square inch pressure weighs 0·1234 lb.

10. What must be the temperature of a normal gas engine mixture if its internal energy is 35200 ft.-lb. per standard cubic foot? Use the formula derived from the Langen formulæ for the constituents.

11. Taking the normal gas engine mixture given in the example in Art. 376, p. 522, and using the Langen formulæ for the specific heats of N_2 and O_2 and H_2O but using the Holborn and Henning formula for CO_2, namely,

$$K_v = 26{\cdot}97 + 0{\cdot}02564t - 0{\cdot}000{,}00933t^2 \text{ ft.-lb. per cubic foot,}$$

determine K_v for the mixture, also the formula for the internal energy. Calculate by means of this formula and also by the formulæ of Langen, Wimperis, and Stodola, the internal energy of the mixture at 500° C., 1000° C., 1500° C., and 2000° C. Tabulate the results.

12. An analysis of the exhaust gases from a gas engine gave, by volume per cent., reduced to standard pressure and temperature, CO_2 5·7, H_2O 12·6, and N_2 and O_2 81·7. Using the Langen formulæ for the specific heats of these gases, determine the formula for the internal energy of this mixture, and use it to calculate the increase in the internal energy for a change of temperature from 230° C. to 1390° C.

13. The exhaust gases from a Diesel engine, working at full load, were found to consist of, CO_2 7 per cent.; CO, N_2, and O_2 together, 87 per cent.; and H_2O 6 per cent., by volume at standard pressure and temperature. By means of the Langen formulæ for the specific heats of these gases, find the formula for the internal energy of this mixture and apply it to find the internal energy at 1000° C.

14. A gas engine having a cylinder 7 inches in diameter and a stroke of 15 inches used 0·0301 cubic foot of gas, at standard pressure and temperature, per working cycle. The ratio of air to gas, by volume, in a working cycle was found to be 8·2. Calculate the volumetric efficiency.

15. The results of four full load tests, by Professors Asakawa and Petavel, of a four-stroke cycle gas engine, are given in the table below. The engine had a cylinder 11 inches in diameter and a stroke of 19 inches. Compute for each test the volume of gas, at standard pressure and temperature, taken in, on the average, in one working cycle, and then calculate the volumetric efficiency.

Test.	(1)	(2)	(3)	(4)
Compression ratio	5·61	4·85	4·11	3·75
Indicated horse-power	31·5	31·6	28·2	31·0
Revolutions per minute	203·8	204·0	200·5	203·3
Percentage of working cycles	84·9	83·6	80·5	93·6
Ratio of air to gas in average working cycles	8·6	7·9	8·3	7·7
Lower calorific value of gas, B.Th.U. per cubic foot, 0° C., 760 mm.	497	493	528	488
Indicated thermal efficiency, per cent.	35·3	33·5	30·5	29·1

16. Determine the volumetric efficiency of a four-stroke cycle gas engine from the following data. Diameter of cylinder, 9 inches. Stroke of piston, 17 inches. Gas used per hour, 285 cubic feet. Pressure of gas, 14·95 lb. per sq. in. Temperature of gas, 17° C. Air used per hour, 2812 cubic feet. Pressure of air, 14·9 lb. per sq. in. Temperature of air, 17° C. Working cycles per hour, 4890. Revolutions per minute, 200. Assume that the pressure in the cylinder at the end of the exhaust stroke, and at the end of the suction stroke, is 14·7 lb. per sq. in. Assume also that the volume of the charge (gas and air in a working cycle, and air only in an idle cycle), in standard cubic feet, is the same for an idle cycle as for a working cycle. Find also the ratio of air to gas, by volume, in a working cycle.

17. A gas engine having a cylinder 10 inches in diameter and a stroke of 18 inches has a volumetric efficiency of 81 per cent. The ratio of air to gas is 8 to 1, and the calorific value of the gas is 275 C.H.U. per cubic foot at standard pressure and temperature. Compute the amount of heat supplied to the engine in a working cycle. Also, if the compression ratio is 4·9, what is the heat value of the mixture per working cycle per cubic foot of total cylinder volume?

18. The volumetric efficiency of a gas engine was found to be 80 per cent. The compression ratio was 5·5. The temperature at the end of the suction stroke was measured directly and was found to be 117° C. Assuming atmospheric pressure in the cylinder at the end of the exhaust stroke and at the end of the suction stroke, compute the temperature at the end of the exhaust stroke.

19. In a gas engine with a compression ratio of 4·8, a volumetric efficiency of 81 per cent., and an exhaust temperature of 450° C., what is the probable suction temperature?

20. A standard cubic foot of normal gas engine mixture, at a pressure of 600 lb. per square inch, and an absolute temperature of 2500° C., expands adiabatically until its absolute temperature is 1600° C. Find the pressure at the end of the expansion, and the expansion ratio. What is the area under the adiabatic in foot-pounds per standard cubic foot of the mixture? Find the value of the index n in $pv^n = C$ which is the equation to a curve starting from the initial point of the true adiabatic and subtending the same base and which has under it the same area as that under the true adiabatic. Take $K_p = 25{\cdot}36 + 0{\cdot}00688T$, and $K_v = 17{\cdot}61 + 0{\cdot}00688T$.

21. A standard cubic foot of mixture in the cylinder of an Otto cycle gas engine contains, before explosion, 9·5 per cent., by volume, of coal gas having a calorific value of 290 C.H.U. per cubic foot. The temperature of the mixture at the end of the suction stroke is 100° C., and its pressure is 14·7 lb. per sq. in. The compression ratio is 4·9. Assume for the compression curve $pv^{1{\cdot}37}$ = constant. Assume a contraction of 2 per cent. in the standard cubic foot of mixture after explosion. The expansion after explosion is adiabatic. Take the Langen formulæ for the specific heats of the mixture after explosion. Compute the temperatures and pressures at the end of the explosion and at the end of the expansion. Find also the ideal efficiency and the air standard efficiency.

22. Taking $K_p = 0{\cdot}228 + 0{\cdot}000{,}06T$, and $K_v = 0{\cdot}158 + 0{\cdot}000{,}06T$ for a gaseous mixture, calculate the changes of entropy for the following changes of state of the mixture, T being in degrees Centigrade absolute, v in cubic foot per lb. of mixture, and p in lb. per sq. in.

(*a*) From T = 460, and v = 11 to T = 1250, and v = 2·8.
(*b*) From T = 1480, and p = 126 to T = 1100, and p = 43.
(*c*) At constant volume from T = 650 to T = 1650.
(*d*) At constant pressure from T = 820 to T = 1760.

23. In a PV diagram the scales are, for pressure, 1 inch to 100 lb. per sq. in., and for volume, 4·25 inches to 16·7 cubic feet. In the Tϕ diagram the scales are, for temperature, 1 inch to 500° C., and for entropy, 2 inches to 0·1 unit. How many ft.-lb. of work are represented by 1 sq. in. of the PV diagram, and how many heat units (C.H.U.) are represented by 1 sq. in. of the Tϕ diagram?

24. The area of the Tϕ diagram for a gas engine, measured with a planimeter, was found to be 11·8 sq. in., and 1 sq. in. of the diagram represented 10 C.H.U. The volume of mixture per lb. in the cylinder at the beginning of the compression stroke was 16·2 cu. ft., and the compression ratio was 6. From these particulars calculate the mean effective pressure in the cylinder during a cycle.

CHAPTER XXIV

PERFORMANCE OF INTERNAL COMBUSTION ENGINES

388. Test of a Gas Engine.—The observations to be made and the method of dealing with them, in order to determine the performance of a gas engine, are indicated in the following example.

Engine dimensions.—The engine, which works on the Otto or four-stroke cycle, has a cylinder 10 inches in diameter and a piston stroke of 18 inches. The clearance volume, measured by filling the clearance space with water, is 353 cubic inches. The brake wheel has an effective diameter of 64 inches.

Duration of trial.—40 minutes.

Observations—Total number of revolutions, 8080. Total number of explosions, 3520. Net load on brake, 202 lb. Mean effective pressure from indicator diagrams, 82 lb. per square inch.

Gas used, 271 cubic feet. Pressure of gas at metre, 5·2 inches of water. Atmospheric temperature, 14° C. Height of barometer, 29·5 inches. Calorific value of gas, by calorimeter, 290 C.H.U. per cubic foot at standard pressure and temperature.

Weight of water passing through jacket, 403·2 lb. Temperature of jacket water at inlet, 14° C., and at outlet, 64° C.

Results.—
$$\text{B.H.P.} = \frac{3{\cdot}1416 \times 64 \times 202 \times 8080}{12 \times 40 \times 33{,}000} = 20{\cdot}72.$$

$$\text{I.H.P.} = \frac{0{\cdot}7854 \times 10^2 \times 82 \times 18 \times 3520}{12 \times 40 \times 33{,}000} = 25{\cdot}76.$$

$$\text{Mechanical efficiency} = \frac{20{\cdot}72 \times 100}{25{\cdot}76} = 80{\cdot}4 \text{ per cent.}$$

Absolute pressure of gas at meter $= 29{\cdot}5 + \dfrac{5{\cdot}2}{13{\cdot}6} = 29{\cdot}88$ inches of mercury, where 13·6 is the specific gravity of mercury.

Absolute temperature of gas at meter $= 273 + 14 = 287°$ C. absolute.

Gas used per minute at standard pressure and temperature

$$= \frac{271}{40} \times \frac{29{\cdot}88}{29{\cdot}92} \times \frac{273}{287} = 6{\cdot}436 \text{ cubic feet.}$$

Gas used per B.H.P. per hour $= \dfrac{6{\cdot}436 \times 60}{20{\cdot}72} = 18{\cdot}6$ cubic feet.

Gas used per I.H.P. per hour $= \frac{6{\cdot}436 \times 60}{25{\cdot}76} = 15{\cdot}0$ cubic feet

Heat converted into work per minute

$$= \frac{25{\cdot}76 \times 33{,}000}{1400} = 607{\cdot}2 \text{ C.H.U.}$$

Heat supplied per minute $= 6{\cdot}436 \times 290 = 1866{\cdot}4$ C.H.U.

Indicated thermal efficiency $= \frac{607{\cdot}2 \times 100}{1866{\cdot}4} = 32{\cdot}5$ per cent.

Heat carried away by jacket water per min. $= \frac{403{\cdot}2}{40} \times (64-14)$ $= 504$ C.H.U. $= \frac{504 \times 100}{1866{\cdot}4} = 27{\cdot}0$ per cent. of the heat supplied.

A heat balance may now be drawn up as follows :—

Heat Balance.

Per minute.	C.H.U.	Per cent.
Heat converted into work	607·2	32·5
Heat carried away by jacket water	504·0	27·0
Heat lost in exhaust gases, by radiation, and unaccounted for (by difference)	755·2	40·5
Heat supplied	1866·4	100·0

The stroke volume is $0{\cdot}7854 \times 10^2 \times 18 = 1413{\cdot}7$ cubic inches.
The clearance volume is 353 cubic inches.

Compression ratio $r = \frac{1413{\cdot}7 + 353}{353} = 5{\cdot}0$.

Air standard efficiency $= 1 - \left(\frac{1}{r}\right)^{1{\cdot}4-1} = 1 - \left(\frac{1}{5}\right)^{0{\cdot}4} = 0{\cdot}475$, or $47{\cdot}5$ per cent.

Relative efficiency $= \frac{32{\cdot}5 \times 100}{47{\cdot}5} = 68{\cdot}4$ per cent.

389. Gas Engine Power Losses and Mechanical Efficiency.—The sources of loss of power in gas engines are—(1) The friction of the mechanism. (2) The air resistance set up by the moving parts. The greatest loss of this kind will generally be the "windage" of the fly-wheel. (3) The pumping losses during the suction and exhaust strokes of the piston, which are represented by the negative loop of the indicator diagram. (4) The loss during compression and expansion in idle cycles, that is, in cycles in which there is no explosion when the engine is governed on the "hit-and-miss" principle. The compression

curve being above the expansion curve the work of compression is not all recovered during expansion.

It is often assumed that the total power loss is constant at all loads in the same engine at constant speed. In a long series of careful and exhaustive tests of a gas engine[1] Professors Asakawa and Petavel found that the power loss at no load was only from 7 per cent. to 10 per cent. less than at full load. The engine used was a 25 brake-horse-power "National" having a cylinder 11 inches in diameter and a stroke of 19 inches. The normal speed of the engine was 200 revolutions per minute and the normal compression ratio 4·85. The details of two tests for power losses, one at full load and the other at no load, at the normal speed and with the normal compression ratio, are given in the following table:—

Power Losses in 11 in. × 19 in. Gas Engine at 200 revs. per min. Compression Ratio, 4·85.

	Full Load.		No Load.	
Friction losses in main bearings, connecting rod, piston, etc. — When engine running without explosion	2·39		2·39	
Friction losses in main bearings, connecting rod, piston, etc. — Additional friction due to increased pressure when exploding	1·65		0·32	
Total		4·04		2·71
Fly-wheel windage loss		0·53		0·53
Pumping losses — Suction	1·30		1·31	
Pumping losses — Exhausting	0·80		1·01	
Total		2·10		2·32
Power loss for compression and expansion in idle cycles		0·11		0·60
Total power losses in horse-power		6·78		6·16
Ratio, working cycles : idle cycles	6 : 1		1 : 6	
Mean effective pressure, calculated lb. per sq. in.	81·3		93·6	
Horse-power — Brake	25		0	
Horse-power — Indicated	31·8		6·2	
Mechanical efficiency per cent.	78·6		0	

Assuming that the relation between the lost power (generally called the friction horse-power) and the brake horse-power is a linear one, which is practically the case, Fig. 639 shows the variation in the friction horse-power (F.H.P.), the indicated horse-power, (I.H.P.), and the mechanical efficiency (M.E.) with variation in the load or brake horse-power.

The equation for the F.H.P. line is $\text{F.H.P} = 0{\cdot}024\text{B} + 6{\cdot}2$

The equation for the I.H.P. line is $\text{I.H.P.} = 1{\cdot}024\text{B} + 6{\cdot}2$

The equation for the M.E. curve is $\text{M.E.} = \dfrac{100\text{B}}{1{\cdot}024\text{B} + 6{\cdot}2}$ per cent.

B = brake horse-power.

[1] "Experimental Investigation of the Thermal Efficiency of a Gas Engine." British Association, September, 1915.

Tests were made with various compression ratios, and the results are represented very approximately by the following formulæ.

Compression ratio.	Power losses.	Compression ratio.	Power losses.
3·75	F.H.P. = 0·0288B + 5·6	4·85	F.H.P. = 0·024B + 6·2
4·11	F.H.P. = 0·024B + 5·9	5·61	F.H.P. = 0·020B + 6·5

From the assumption of a linear relation between the friction horse-power and the brake horse-power, it follows that if in any gas engine it is found that the indicated horse-power is I_0 at no load and I_1 at some other load when the brake horse-power is B_1, then at any brake load B the friction horse-power is $F = \left(\frac{I_1 - I_0 - B_1}{B_1}\right)B + I_0$, and the indicated horse-power is $I = B + F$. The speed of the engine is assumed to be the same throughout.

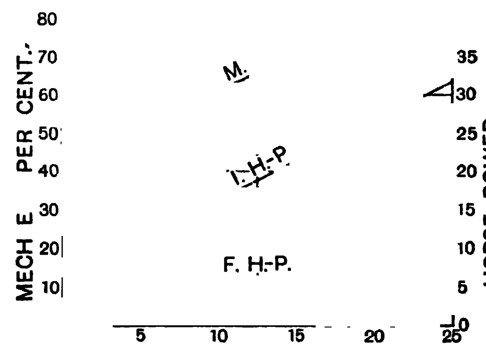


FIG. 639.

It may be mentioned here that the object which Asakawa and Petavel had in view, in determining accurately the power losses in their gas engine tests, was the ultimate determination of the indicated horse-power in a more reliable manner than was possible from indicator diagrams, although they also took indicator diagrams. The brake horse-power can be determined with a high degree of accuracy, and if the lost horse-power is known the correct indicated horse-power is the sum of these two. The methods adopted by Asakawa and Petavel in determining the power losses are fully described in their paper already referred to.

The mechanical efficiency of a gas engine at full load and at the normal speed for which the engine is designed may be taken at from 75 per cent. to 85 per cent., with an average of about 80 per cent. It diminishes slightly with an increase in the compression ratio. It increases as the speed is reduced below the normal speed and decreases as the speed is increased above the normal. In tests by Gibson and Walker[1] at speeds of 150, 200, and 250 revolutions per minute at full load, the mechanical efficiencies were, 86, 81, and 74 per cent. respectively, 200 revolutions per minute being the normal speed of the engine.

390. Thermal Efficiency of Gas Engines.—The thermal efficiency of a gas engine is influenced chiefly by the compression ratio and the strength of the charge, being higher the greater the compression ratio

[1] "The Distribution of Heat in the Cylinder of a Gas Engine," *Proceedings of the Institution of Mechanical Engineers*, May, 1915.

with a given mixture, but with a given compression ratio the strongest mixture does not give the highest efficiency.

The effect of the compression ratio and the strength of the charge is very clearly shown in Fig. 640, which represents the results of the tests, at full load, by Asakawa and Petavel, which have been referred to in the preceding Art. The strength of the charge is indicated by the amount of excess air which it contains.

Excess air

$$= \frac{\text{Total amount of air used} - \text{Air required for complete combustion}}{\text{Air required for complete combustion}}$$

That is, excess air is the ratio of the quantity of air present in the mixture, but not required for complete combustion, to the air required for complete combustion. The strength of an explosive mixture is better indicated by the excess air than by the more usual method of

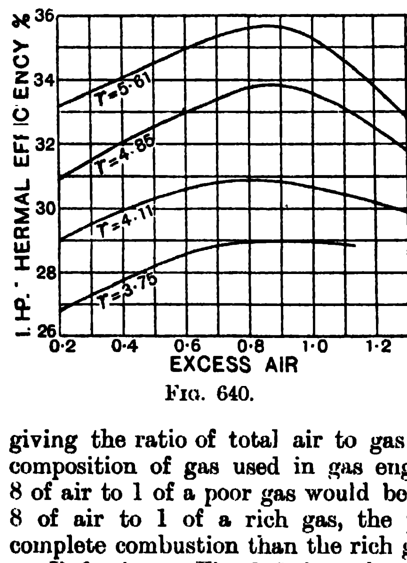


Fig. 640.

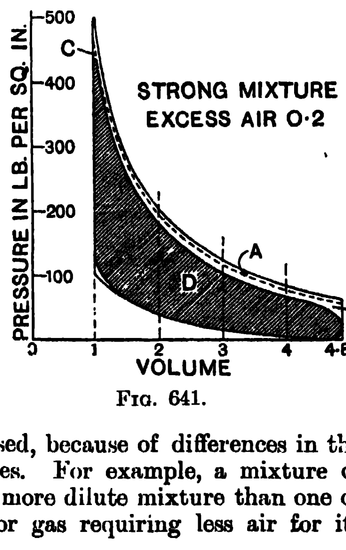


Fig. 641.

giving the ratio of total air to gas used, because of differences in the composition of gas used in gas engines. For example, a mixture of 8 of air to 1 of a poor gas would be a more dilute mixture than one of 8 of air to 1 of a rich gas, the poor gas requiring less air for its complete combustion than the rich gas.

Referring to Fig. 640, it is clearly seen that for a given strength of mixture, as shown by the excess air present, the thermal efficiency is higher the greater the compression ratio. This, of course, would be expected from theoretical considerations (Art. 370). Another result of great importance is shown by the curves in Fig. 640, namely, for all compression ratios the highest efficiency is obtained when the excess air is about 0·8 of that required for complete combustion. Also there is not much change in the efficiency when the excess air varies between 0·7 and 1·0.

It will be seen that with compression ratios of 3·75, 4·11, 4·85, and 5·61, the maximum indicated thermal efficiencies are about 29 per cent., 30·8 per cent., 33·8 per cent., and 35·6 per cent. respectively.

Asakawa and Petavel also found that for all compression ratios between 3·75 and 5·61, and for all mixtures containing between 0 and

0·7 excess air, the efficiency ratio was about 88 per cent. and that for weaker mixtures this ratio falls off rapidly. They also constructed the ideal diagram round the actual indicator diagram and obtained instructive figures as to the nature of the losses. Figs. 641, 642, and 643 show the actual and ideal diagrams for strong, medium, and weak mixtures respectively, at full load and at a speed of 200 revolutions per minute with the normal compression ratio of 4·85. In each case the outer diagram is the ideal diagram for the actual mixture used, allowing for the variation of the specific heat with change of temperature. The inner hatched area is the actual indicator diagram. The dotted curve is the adiabatic which touches the expansion curve of the actual diagram.

The area A between the expansion curve of the ideal diagram and the dotted adiabatic represents the effect of the loss of heat at the beginning of the stroke. The area B between the dotted adiabatic and the expansion curve of the actual diagram represents the loss of work due either to leakage or to the transmission of heat to the cylinder surface during expansion. The area C between the dotted adiabatic

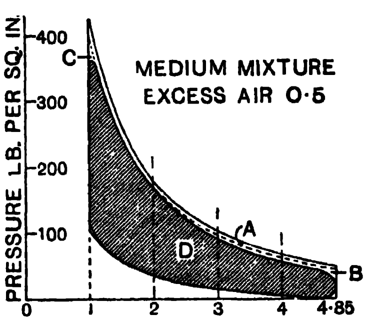


FIG. 642.

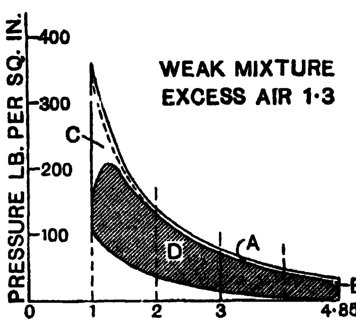


FIG. 643.

and the upper part of the actual diagram represents the loss of work due to delayed explosion.

The values of these areas for strong, medium, and weak mixtures are given by Asakawa and Petavel in the following table. The figures refer to tests at full load at 200 revolutions per minute with a compression ratio of 4·85. The total available energy, as represented by the area of the ideal diagram, is denoted by 100 in each case, hence the numbers given in line (D) are the percentage efficiency ratios.

		Strong	Medium	Weak
Strength of mixture				
Excess air . . .		0·2	0·5	1·3
Waste of energy due to thermal losses	Before expansion . . . (A)	5·0	4·66	9·67
	During expansion . . . (B)	7·1	4·47	0·46
	During explosion . . (C)	0·1	0·68	9·70
	Total thermal losses	12·2	9·81	19·83
Indicated work	. . . . (D)	87·8	90·19	80·17
	Total available energy	100·0	100·00	100·00

If the heat equivalent of the brake horse-power be divided by the heat supplied by the fuel the result is the *B.H.P. thermal efficiency*. This efficiency is of course less than the thermal efficiency as determined from the indicator diagram or from the I.H.P. Asakawa and Petavel plotted the B.H.P. thermal efficiency for various compression ratios and strengths of mixture, as was done for the I.H.P. thermal efficiency (Fig. 640), and similar curves and results were obtained, except of course that the curves were lower. That is to say, they found that for a given strength of mixture the B.H.P. thermal efficiency was higher the greater the compression ratio, and that at all compressions the highest efficiency is obtained when the excess air is in the neighbourhood of 0·8. With compression ratios of 3·75, 4·11, 4·85, and 5·61 the maximum B.H.P. thermal efficiencies were about 23 per cent., 24 per cent., 26·2 per cent., and 27·4 per cent. respectively.

391. Heat Losses in Gas Engines.—Of the heat supplied to a gas engine, the part which is not converted into work in the cylinder is carried off: (*a*) By the cooling water surrounding the cylinder and also, in the case of large engines, by the water used for cooling the piston and valves; (*b*) by the exhaust gases; and (*c*) by radiation from the various hot parts of the engine.

(*a*) The cooling water usually enters at about the temperature of the atmosphere, and its rate of flow is regulated so that its temperature at exit is from 40° C. (104° F.) to 75° C. (167° F.).

If w is the weight in lb. of the water passing through the jacket in a given time and t its average rise in temperature, then the heat carried off by the cooling water in that time is wt heat units.

At full load under ordinary conditions the cooling water carries away from 25 per cent. to 33 per cent. of the heat supplied to the engine, an average value being about 28 per cent. Probably one-third of this amount is taken from the exhaust gases during the exhaust stroke, and should properly be added to the amount usually credited to the exhaust gases.

(*b*) At full load under ordinary conditions the amount of heat in the exhaust gases on leaving the engine varies from 35 per cent. to 45 per cent. of the heat supplied to the engine and may be taken on the average as 40 per cent.

In tests of gas engines the total heat supplied (H), the heat converted into work (I), and the heat carried away by the cooling water (C) are readily determined, and the amount H − I − C is then frequently given as the heat carried away by the exhaust gases and unaccounted for, and is not measured directly.

The heat in the exhaust gases may, however, be determined if their temperature near the exhaust valve be taken and their weight and specific heat be found. The weight of the exhaust gases is equal to the weight of air and gas entering the engine, and their mean specific heat may be computed from an analysis and a knowledge of the specific heats of the constituents.

The heat in the exhaust gases may also be measured directly by passing them through a calorimeter. In a surface calorimeter the gases pass through or around a number of tubes while the cooling

water passes around or through the tubes. In a jet calorimeter the gases are brought into direct contact with jets of cooling water.

(*c*) Radiation losses are difficult to determine directly and the amount of heat loss credited to radiation by different experiments varies greatly. Burstall[1] found the radiation loss to be about 3 per cent. of the total heat supplied, with a jacket temperature of about 63° C. Hopkinson[2] estimated the radiation loss as not exceeding 4 per cent. with a jacket temperature of 70° C. Gibson and Walker[3] gave the amount as 8 per cent. at full load under normal conditions with a jacket temperature of 63° C.

392. Gas Engine Average Heat Balance.—Working at full load under normal conditions the performance of a gas engine, as shown by a heat balance-sheet, should not differ much from that shown in the following table.

Average Heat Balance for a Gas Engine.

Heat converted into work as shown by indicator diagram	33
Heat carried away by the cooling water *	27
Heat carried away by exhaust gases and unaccounted for	40
Total heat supplied	100

* Probably about one-third of this should be credited to the exhaust gases.

393. Gas Consumption and Power—Willans Line.—For a steam engine, governed by throttling and running at constant speed, it has been shown (Art. 199, p. 277) that if the total steam consumption be plotted on a horse-power base the points obtained lie on a straight line called the *Willans line.* It will be found that for a gas engine running at approximately constant speed, if the gas consumption be plotted on a horse-power base, the points obtained also lie very approximately on a straight line, which is the Willans line for the engine. An example is illustrated in Fig. 644. The actual total gas consumption at various brake loads has been plotted and the straight line which most nearly contains all the points has been drawn. The equation to this line is $G = 12{\cdot}8B + 158$, where G is the total gas consumption in cubic feet per hour, and B is the corresponding brake horse-power. The number 158 is obviously the consumption of gas at no load. G′, the

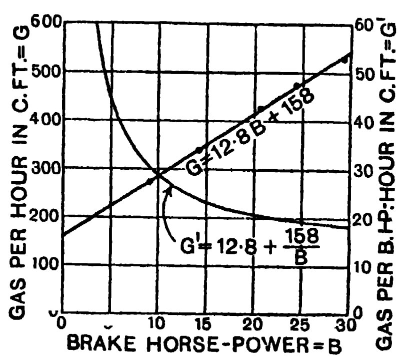


FIG. 644.

[1] *Proceedings of the Institution of Mechanical Engineers*, October, 1901.
[2] " " " " " April, 1908.
[3] " " " " " May, 1915.

consumption of gas in cubic feet per brake horse-power per hour, has also been plotted. The equation to the resulting curve is

$$\frac{G}{B} = G' = 12 \cdot 8 + \frac{158}{B}$$

In reporting on the performance of a gas engine, the total gas consumption in cubic feet per hour and also the consumption per brake horse-power and per indicated horse-power per hour are generally stated, but as a measure of the efficiency of the engine the figures for gas consumption may be misleading. For example, a consumption of 18 cubic feet per brake horse-power per hour of a gas having a calorific value of 290 C.H.U. would be equivalent to a consumption of 22 cubic feet of a gas having a calorific value of 237 C.H.U.

394. Performance of Petrol Engines.—Petrol engines are used mainly for motor vehicles, aeroplanes, and airships, and, when working at full power, run usually at from 1000 to 2000 revolutions per minute. Engines running at 3500 revolutions per minute have been used on racing motor cars.

Mechanical Efficiency.—Working at full power the mechanical efficiency is usually not less than 80 per cent. and at reduced speed it may amount to 88 per cent.

Mean Effective Pressure.—Working in the neighbourhood of full power the M E P. in the cylinder usually varies from 90 to 120 lb. per square inch. The M E.P. equivalent of the B.H P. may be taken as varying from 70 to 100 lb per square inch.

Fuel Consumption and Thermal Efficiency.—Working in the neighbourhood of full power the consumption of petrol may be as low as 0·47 lb. per B.H.P. per hour, which corresponds to a brake thermal efficiency of about 29 per cent., and an indicated thermal efficiency of about 36 per cent., if the mechanical efficiency is 80 per cent.

More commonly the best performance of a petrol engine is a consumption of about 0·65 lb. per B.H.P. per hour which corresponds to a brake thermal efficiency of about 21 per cent., and an indicated thermal efficiency of about 26 per cent. if the mechanical efficiency is 80 per cent.

395. Tests of a Motor Car Engine.—In May, 1909, a paper by Professor W. Watson was read before the Institution of Automobile Engineers on "The Thermal and Combustion Efficiency of a Four-Cylinder Petrol Motor," which gave the results of a very complete series of tests which are worthy of careful study. The motor tested was a Clement-Talbot motor car engine having four cylinders 85 mm. (3·35 in.) in diameter and a piston stroke of 120 mm. (4·72 in.).

The engine, as designed by the makers, had a compression ratio of 4·03. Three different compression ratios were used, and with each compression ratio three series of tests were made at different speeds. The compression ratios, and the highest speed in each of the series of tests, are given in the following table, which also gives the compression pressures at various speeds, the falling off in pressure being of course due to increased wire-drawing action with increase in speed.

Test Letter.	Compression Ratio.	Highest Speed in revs. per min.			Compression Pressure (G in lb. per sq. in. at		
		Series. 1	Series. 2	Series. 3	720 R.P.M.	1090 R.P.M.	1270 R.P.M.
A	4·71	1290	1111	723	88	85	83
B	4·35	1274	1103	720	80	77	75
C	3·92	1255	1087	715	72	68	66

In each series of tests the ratio of air to petrol was varied.

The results of the tests A, series 1, have been plotted and the curves shown in Figs. 645, 646, and 647 represent very approximately the performance of the engine under the conditions of these tests. The most prominent lesson which these curves teach is that the

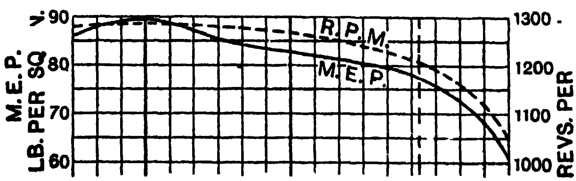


FIG. 645.

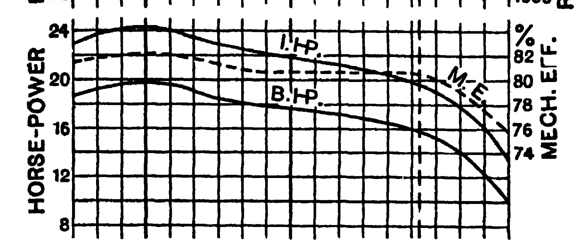


FIG. 646.

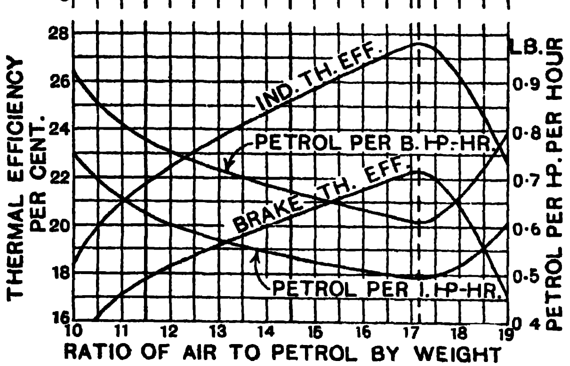


FIG. 647.

mixture of air and petrol which gives the greatest power is not the same as that which gives the highest thermal efficiency, and neither of these mixtures is the one for complete combustion with the minimum supply of air. The mixture for complete combustion without excess air may be taken as 14·5 of air to 1 of petrol, by weight, and a thick

vertical line is shown through this point. For all mixtures to the right of this 14·5 line the air is in excess, while for all mixtures to the left there is deficiency of air and combustion must be incomplete.

Fig. 647 shows that in the tests now under consideration the most economical mixture is one containing rather more than 17 of air to 1 of petrol, by weight, so that the excess air for highest thermal efficiency is roughly 20 per cent. In the case of the gas engine tests referred to in Art. 390, it was shown that the mixture for highest thermal efficiency contained about 80 per cent. excess air.

Referring to Fig. 646 it is seen that the greatest power is developed with a mixture of about 11·5 of air to 1 of petrol, by weight, that is, a mixture in which the amount of air is only sufficient for the complete combustion of about 80 per cent. of the petrol present. The result is that the products of combustion contain combustible gases, such as carbonic oxide (CO), hydrogen (H), and marsh gas (CH_4) and a lowering of the thermal efficiency must of course follow. When, however, petrol vapour is burned with a supply of air insufficient for complete combustion, the volume of the products is greater than the original volume of the mixture, at the same pressure and temperature, to a larger extent the smaller the amount of air present in the original mixture. This is shown by the diagram in Fig. 648 due to Professor Watson. Hence this tendency of the products to occupy a greater volume, apart from the effort to expand due to the heat produced, will cause a higher pressure in the cylinder of the engine, increasing the mean effective pressure on the piston, and consequently increasing the power. This accounts, at least partly if not altogether, for the maximum power being developed with an excess of petrol in the mixture. With too great an excess of petrol, however, the pressure will be lower because of the falling off in the supply of heat.

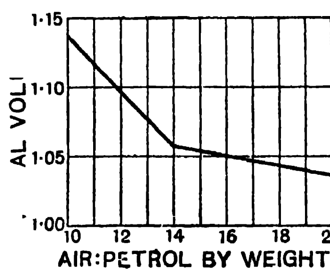


Fig. 648.

The influence of speed and compression ratio on the maximum power and maximum thermal efficiency, and on the ratio of air to petrol giving maximum power or maximum thermal efficiency, may be studied by reference to the table on the next page, the figures in which are taken direct from Professor Watson's paper, except that where two or more powers or thermal efficiences in the neighbourhood of maximum power or maximum thermal efficiency are nearly equal the mean of the results has been taken in each case.

It will be noticed that the improvement in the performance of the engine due to increasing the compression ratio is not very marked.

It will also be seen that with all compression ratios and at all speeds there is a marked diminution in the power developed when the carburettor is set for maximum economy instead of for maximum power.

Compression Ratio	Series 1. Revolutions per minute.	Series 1. Indicated Horse-power.	Series 1. Thermal Efficiency, per cent.	Series 1. Air : Petrol by weight.	Series 2. Revolutions per minute.	Series 2. Indicated Horse-power.	Series 2. Thermal Efficiency, per cent.	Series 2. Air : Petrol by weight.	Series 3. Revolutions per minute.	Series 3. Indicated Horse-power.	Series 3. Thermal Efficiency, per cent.	Series 3. Air : Petrol by weight.
						Maximum Power.						
4·71	1287	**24·2**	22·2	11·6	1108	**20·6**	21·0	12·0	721	**13·4**	23·1	14·8
4·35	1265	**22·9**	21·8	11·8	1103	**20·2**	22·1	12·7	720	**13·3**	19·8	12·8
3·92	1251	**21·8**	20·8	11·9	1087	**19·5**	20·8	12·3	715	**13·1**	20·3	13·1
						Maximum Thermal Efficiency.						
4·71	1211	19·8	**27·5**	17·2	1041	17·2	**26·1**	17·2	674	11·2	**24·8**	18·0
4·35	1195	19·1	**27·4**	17·4	1027	16·6	**26·4**	17·7	655	10·4	**24·9**	18·9
3·92	1213	20·5	**26·8**	16·0	1032	17·3	**25·0**	16·2	655	10·4	**23·7**	17·8

396. Performance of a Petrol Engine at Various High Speeds.—In an instructive paper on "High-speed Internal Combustion Engines," read before the North-East Coast Institution of Engineers and Shipbuilders in 1918, Mr. Harry B. Ricardo plotted the probable performance of a high-speed petrol engine at speeds varying from 400 to 2400 revolutions per minute. The curves obtained are shown in Figs. 649 and 650. The leading dimensions of the engine to which these curves refer are as follows.—

Diameter of cylinder	7·25 in.
Stroke of piston	8·50 in.
Weight of reciprocating mass	11·35 lb.
Weight of rotating mass	8·00 lb.
Number of valves	2 inlet and 2 exhaust
Diameter of valve ports	2·5 in.
Lift of valves	0·65 in.
Ratio of piston area to effective inlet port area	3·94 : 1
Diameter of crank pin	3·25 in.
Width of crank pin	3·75 in.

The range of speed, 400 to 2400 revolutions per minute, corresponds, to a range of mean piston speed, 567 to 3400 feet per minute.

Mr. Ricardo assumed a compression ratio of 5 : 1, which he stated was just possible, provided that every part of the combustion chamber, and particularly the surroundings of the exhaust valves and sparking plugs, are well cooled by water in sufficiently rapid circulation to break away any steam bubbles that may be formed. With a compression ratio of 5 : 1 the air standard efficiency is 47·5 per cent.

Volumetric efficiency. (*a*) Fig. 649.—The highest volumetric efficiency is about 79 per cent. and occurs at a speed of about 1000 revolutions per minute. At lower speeds the falling off of the volumetric efficiency is due to the valves being set to give the best efficiency at the higher speeds and, in consequence, the inlet valves are held open too late at the lower speeds, with the result that a small portion of the charge is rejected through them. As the speed increases above that giving the

highest volumetric efficiency that efficiency falls off because of wire-drawing.

Mean Effective Pressure. (*b*) Fig. 649.—The indicated mean effective pressure (I.M.E.P.) is calculated from the formula

$$P_m = E_v \times \frac{C \times 778}{144} \times E_t$$

where E_v is the volumetric efficiency, E_t the indicated thermal efficiency, and C the calorific value of 1 cubic foot of the *mixture* in the

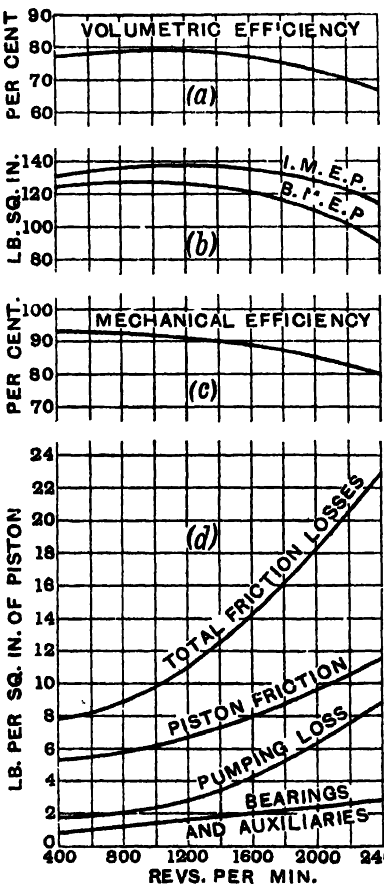


Fig. 649.

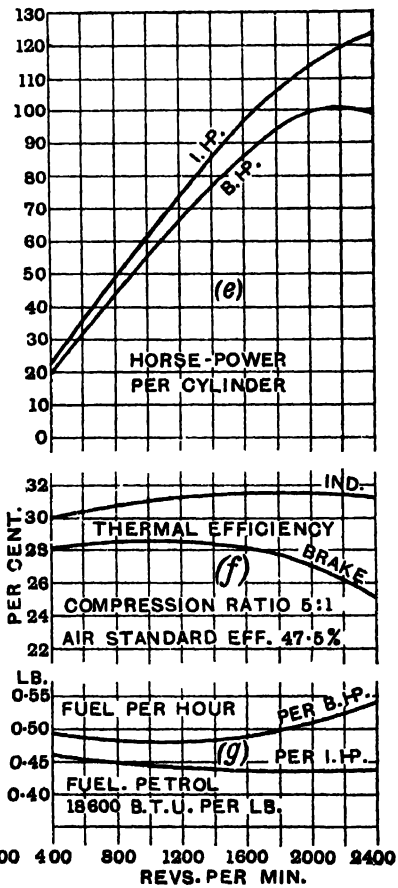


Fig. 650.

cylinder before compression at standard pressure and temperature in B.Th.U. The value of C is taken at 98 B.Th.U.

The brake mean effective pressure (B.M.E.P.) is obtained by deducting the pressure required to overcome the frictional resistances.

Mechanical Efficiency. (*c*) Fig. 649.—This varies from about 94 per cent. at the lowest speed to 80 per cent. at the highest speed.

Friction Losses. (*d*) Fig. 649.—The friction losses are made up of—(1) the friction of the bearings and the resistances due to auxiliaries including a centrifugal water circulating pump; (2) the pumping loss during the suction and exhaust strokes; and, (3) the piston friction. The total friction losses reach the equivalent of about 23 lb. per square inch of piston at the highest speed.

Horse-power. (*e*) Fig. 650.—The brake horse-power reaches a maximum of 100 at a speed of 2100 revolutions per minute.

Thermal Efficiency. (*f*) Fig. 650.—The indicated thermal efficiency varies from 30 per cent. at the lowest speed to a maximum of about 31·5 per cent. at about 2000 revolutions per minute, beyond which it diminishes slightly. The brake thermal efficiency varies from about 28·1 per cent. at the lowest speed to a maximum of about 28·7 per cent. at 1000 revolutions per minute, beyond which it falls off, at first slowly, and then more rapidly, to about 25 per cent. at the highest speed.

Fuel per Horse-power per Hour. (*g*) Fig. 650.—The weight of petrol used per indicated horse-power per hour is about 0·46 lb. at the lowest speed and diminishes gradually with increase of speed to a minimum of about 0·44 lb. On the brake horse-power the minimum consumption is about 0·48 lb. at 1000 revolutions per minute and reaches a maximum of about 0·54 lb. at the highest speed.

NOTE.—The foregoing efficiencies estimated, or assumed, by Mr. Ricardo are mostly higher than have hitherto been common, but the compression ratio of 5 which he assumes, and upon which his calculations are based, is higher than is usual in petrol engines.

397. Performance of Diesel Engines.—The following notes and observations are based on a study of the results of a considerable number of tests of Diesel engines of all sizes but most of them over 100 B.H.P.

Mean Effective Pressure.—At full load the M.E.P. from the indicator diagrams varies in practice from 85 to 120 lb. per square inch for both four-stroke and two-stroke cycle engines. The average value of the M.E.P. with varying loads at constant speed is given by the formula—

$$\text{Mean effective pressure} = 23 + 80x \text{ lb. per square inch,}$$

where x is the ratio which the load bears to the full brake load for which the engine is designed. For example: at full load $x = 1$, at half load $x = 0·5$, and for an overload of 10 per cent. $x = 1·1$.

Mechanical Efficiency.—In four-stroke cycle engines running at normal speed and full load the mechanical efficiency varies in general from 75 to 80 per cent.

The difference between the I.H.P. and the B.H.P. is about the same at all loads at normal speed, and on the average the mechanical efficiency at any load is given by the formula.—

$$\left.\begin{array}{r}\text{Mechanical efficiency of}\\ \text{four-stroke cycle engines}\end{array}\right\} = \frac{100x}{x + 0·29} \text{ per cent.}$$

where x has the same meaning as before.

Since the two-stroke cycle engine has to be provided with a scavenging pump, which absorbs from 5 to 10 per cent. of the I.H.P. of the engine, it follows that under similar conditions of lubrication,

amount of compression, etc., its mechanical efficiency will be less than that of the four-stroke cycle engine. The average mechanical efficiency at any load and at normal speed may be taken at about that given by the formula—

$$\left.\begin{array}{r}\text{Mechanical efficiency of}\\ \text{two-stroke cycle engines}\end{array}\right\} = \frac{100x}{1{\cdot}03x + 0{\cdot}3} \text{ per cent.}$$

At speeds higher than the normal speed the mechanical efficiency of both types of engine falls off, and at speeds below the normal it increases slightly, the mean effective pressure being kept constant.

Indicated Thermal Efficiency.—At full load and normal speed the indicated thermal efficiency varies in general from 38 to 44 per cent., but an indicated thermal efficiency as high as 48 per cent. at full load has been claimed.

If W is the weight of fuel used by the engine in lb. per hour, and C is the calorific value of the fuel in C.H.U. per lb. then,

$$\text{Indicated thermal efficiency} = \frac{33{,}000 \times 60 \times \text{I.H.P.} \times 100}{1400 \times \text{W} \times \text{C}} \text{ per cent.}$$

Under the conditions prevailing in Diesel engines the combustion of the fuel is fairly complete at an early part of the stroke at all ordinary loads, but, as the excess air increases as the load is reduced, the temperature must diminish, and the loss of heat through the cylinder walls will therefore decrease. The consequence is that the thermal efficiency increases as the load diminishes.[1] Plotting the indicated thermal efficiency at varying loads on a load base it is found, from the results of actual tests, that the points are practically in a straight line and the equation to this line, for both four-stroke and two-stroke cycle engines, may be taken as—

$$\text{Indicated thermal efficiency} = 52 - 11x \text{ per cent.}$$

With constant M.E.P. a moderate variation in speed has very little influence on the indicated thermal efficiency but it generally falls off slightly as the speed increases.

Net Indicated Horse-Power and True Thermal Efficiency.—The I.H.P. of the air compressor which supplies the air for the injection of the fuel may be taken at from 6 to 8 per cent. of the I.H.P. of the engine, and, neglecting loss, this is returned to the cylinder of the engine. The difference between the I.H.P. shown by the indicator diagram and the I.H.P. returned to the cylinder from the compressor is called the *net* I.H.P. of the engine. In computing the net I.H.P. in practice it is usual to neglect any loss of power in the air as it passes from the compressor to the engine cylinder.

The horse-power developed in the cylinder due to the combustion of the fuel is the net I.H.P. as defined above and the *true* thermal efficiency is that based on the net I.H.P. For example, let the indicated thermal efficiency be 42 per cent., and let the net I.H.P. be 94 per cent. of the diagram I.H.P., then the true thermal efficiency is $42 \times 0{\cdot}94 = 39{\cdot}48$ per cent.

If the fuel consumption is 0·335 lb. per I.H.P. per hour and the

[1] Theory (Art. 374) also shows that the efficiency is higher the greater the expansion ratio.

net I.H.P. is 94 per cent. of the diagram I.H.P., then the fuel consumption per net I.H.P. per hour is $0{\cdot}335 \div 0{\cdot}94 = 0{\cdot}356$ lb.

Brake Thermal Efficiency.—At full load and normal speed the brake thermal efficiency varies in general from 28 to 33 per cent., but a brake thermal efficiency as high as 38 per cent. has been claimed.

The brake thermal efficiency is equal to the product of the mechanical efficiency and the indicated thermal efficiency. Hence, using the formulæ already given for mechanical efficiency and indicated thermal efficiency, the following formulæ are obtained—

$$\left.\begin{array}{r}\text{Brake thermal efficiency of}\\ \text{four-stroke cycle engines}\end{array}\right\} = \frac{x(52 - 11x)}{x + 0{\cdot}29} \text{ per cent.}$$

$$\left.\begin{array}{r}\text{Brake thermal efficiency of}\\ \text{two-stroke cycle engines}\end{array}\right\} = \frac{x(52 - 11x)}{1{\cdot}03x + 0{\cdot}3} \text{ per cent.}$$

For either of these formulæ the brake thermal efficiency is a maximum when $x = 0{\cdot}92$. This is obtained in the usual way by differentiating the expression and equating the result to zero.

The results of tests often show that the highest brake thermal efficiency, and therefore the lowest fuel consumption per B.H.P. per hour, occurs at between three-quarter load and full load.

If W is the weight of fuel used by the engine in lb. per hour, and C is its calorific value in C.H.U. per lb., then,

$$\text{Brake thermal efficiency} = \frac{33{,}000 \times 60 \times \text{B.H.P.} \times 100}{1400 \times W \times C} \text{ per cent.}$$

Fuel Per Horse-Power Per Hour.—W and C stand for the same quantities as above.

$$\left.\begin{array}{r}\text{Fuel in lb. per I.H.P.}\\ \text{per hour} \quad . \quad . \quad .\end{array}\right\} = \frac{W}{\text{I.H.P.}} = \frac{33{,}000 \times 60}{1400 \times C \times \text{Ind. Th. Eff.}}$$

$$\left.\begin{array}{r}\text{Fuel in lb. per net}\\ \text{I.H.P. per hour}\end{array}\right\} = \frac{W}{\text{net I.H.P.}} = \frac{33{,}000 \times 60}{1400 \times C \times \text{True Th. Eff.}}$$

$$\left.\begin{array}{r}\text{Fuel in lb. per B.H.P.}\\ \text{per hour} \quad . \quad . \quad .\end{array}\right\} = \frac{W}{\text{B.H.P.}} = \frac{33{,}000 \times 60}{1400 \times C \times \text{Brake Th. Eff.}}$$

It is necessary, in examining figures of fuel consumption, to know the calorific value of the fuel used, otherwise the figures may be misleading. For example, a consumption of 0·425 lb. of oil having a calorific value of 10,400 C.H.U. (18,720 B.Th.U.) per lb. would be equivalent to a consumption of 0·442 lb. of oil having a calorific value of 10,000 C.H.U. (18,000 B.Th.U.) per lb.

The results tabulated on next page have been calculated by means of the formulæ which have been given, which make the results plot into straight lines or smooth curves as shown in Figs. 651 and 652.

Average Performance of a Four-Stroke Diesel Engine at Constant Speed.

Load	0·25	0·5	0·75	0·92	Full	1·1
M.E.P. . . lb. per sq. in.	43·0	63·0	83·0	96·6	103·0	111·0
Mechanical efficiency per cent.	46·3	63·3	72·1	76·0	77·5	79·1
Indicated thermal eff. ,,	49·2	46·5	43·7	41·9	41·0	39·9
True thermal eff. * ,,	46·3	43·7	41·1	39·4	38·5	37·5
Brake thermal eff. ,,	22·8	29·4	31·6	31·84	31·78	31·6
Fuel per I.H.P. per hour † lb.	0·287	0·304	0·323	0·338	0·345	0·354
Fuel per net I.H.P. per hour † lb.	0·305	0·324	0·344	0·359	0·367	0·377
Fuel per B.H.P. per hour † lb.	0·620	0·481	0·448	0·444	0·445	0·448
Fuel per hr. for full load of 100 B.H.P. † . . . . . lb.	15·5	24·0	33·6	40·8	44·5	49·3

* Assuming that the net I.H.P. is 94 per cent. of the diagram I.H.P.
† Taking the calorific value of the fuel as 10,000 C.H.U. (18,000 B.Th.U.).

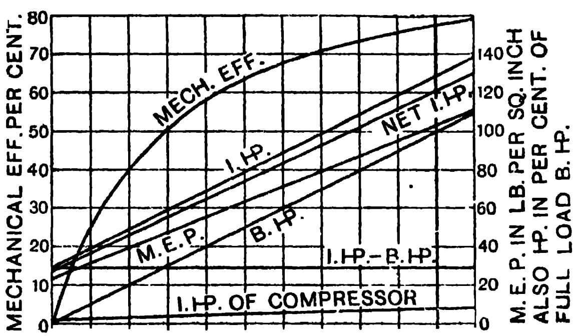


FIG. 651.

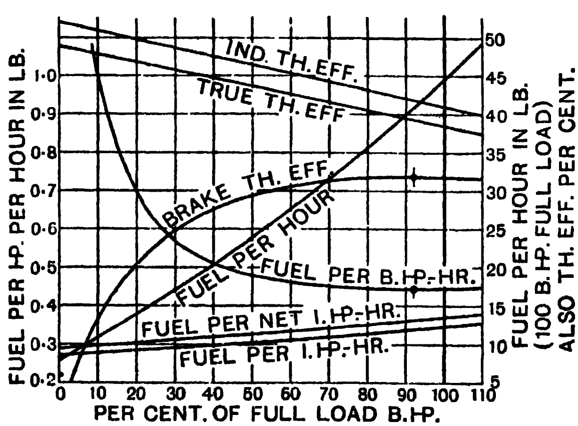


FIG. 652.

Examining Fig. 652 it will be noticed that the fuel per hour curve is very flat, showing that it approximates to a Willans line. The

curves for fuel per I.H.P. per hour, and per net I.H.P. per hour, are very nearly straight lines. The highest point in the brake thermal efficiency curve and the lowest point in the fuel per B.H.P. per hour curve are marked by dots.

Heat Balance.—The heat carried away by the cooling water varies in general from 20 to 32 per cent. of the heat supplied, the average being about 27 per cent. Taking this average value, and taking the values of the *true* thermal efficiency from the table already given, the approximate heat balance for various loads would be as in the following table

Diesel Engine Heat Balance, Per Cent.

Load x	0·25	0·5	0·75	Full	1·1
Heat converted into work	46	44	41	39	38
Heat carried away by cooling water . .	27	27	27	27	27
Heat carried away in exhaust gases, by radiation, and unaccounted for . . .	27	29	32	34	35
Heat supplied .	100	100	100	100	100

Temperature of Exhaust Gases.—At normal speed the temperature of the exhaust gases at different loads is, on the average, about as given in the following table.

Average Temperature of Exhaust Gases of Diesel Engines.

Load x	0·25	0·5	0·75	Full	1·1
Four-stroke cycle engines . . . . °C.	185	240	300	390	420
Two-stroke cycle engines . . . . . . °C.	160	190	230	270	290

It will be observed that for two-stroke cycle engines the temperatures are considerably lower than for four-stroke cycle engines. This is due to the cooling action of the scavenging air. At constant load the temperature of the exhaust gases rises as the speed of the engine increases.

398. Tests of a Crossley Oil Engine.—In 1918 Prof. F. W. Burstall made numerous tests of the latest form of the Crossley heavy oil engine, the results of which are of interest. The engine was of the horizontal type having a cylinder 18·5 inches in diameter and a stroke of 28 inches. The normal rated load was 117 B.H.P. at 180 revolutions per minute. The engine was self-igniting and started cold with compressed air at a pressure of 130 lb. per square inch.

Three series of tests were made, one series with kerosene having a calorific value of 18,500 B.Th.U. per lb., one with residual petroleum having a calorific value of 18,000 B.Th.U. per lb., and one with tar oil

having a calorific value of 16,200 B.Th.U. per lb. The calorific values were determined by means of a Berthelot bomb calorimeter.

The results of the tests with residual petroleum are plotted in Fig. 653 and summarized in the table below.

Brake horse-power	142·0 *	125·0	96·7	30·9
Oil per B.H.P. per hour . . . . lb.	0·457	0·425	0·424	0·647
Heat units per B.H.P. per hour B.Th.U.	8226	7650	7632	11646
Thermal efficiency on B.H.P. per cent.	30·93	33·26	33·35	21·85

* 21 per cent. overload.

At no load the oil fuel consumption was 11·0 lb. per hour.

At the end of compression, quoting from Prof. Burstall's report, when ignition commences, the pressure rises very considerably in a similar manner to that in a gas engine; it then commences to fall although the charge is still burning, but the maximum temperature has not been reached due no doubt to the fact that the oil injection is

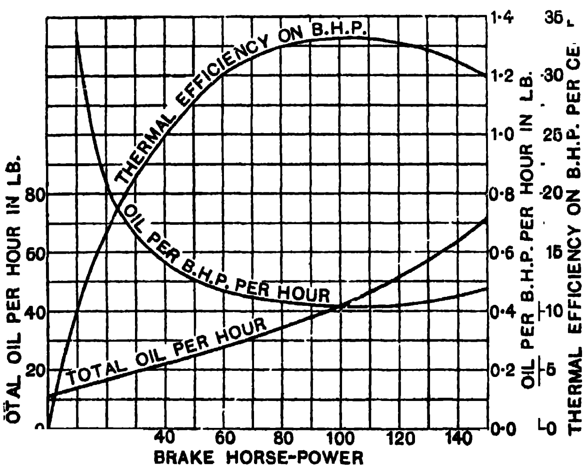


FIG. 653.

still continuing. When the piston has moved through a quarter of its stroke, maximum temperature is reached and from thence to nearly two-thirds along the stroke the expansion is nearly isothermal. From two-thirds of the stroke to the opening of the exhaust valve the temperature is falling but is still above what it would be had the expansion been adiabatic.

The temperature of the exhaust followed closely the power of the engine, ranging from 100° C. at light load to 465° C. on the overload.

399. Performance of Low and Medium Compression Oil Engines.—The performance of low and medium compression oil engines working at full load and normal speed is shown by the typical results of tests tabulated on the next page.

Performance of Low and Medium Compression Oil Engines.

Cycle.		Four-stroke.				Two-stroke*
Compression pressure	lb. per sq. in.	50	60	169	260	185
Maximum pressure	lb. per sq. in.	230	168	325	475	300
Mean effective pressure	lb. per sq. in.	68·5	48	75	82	49
Mechanical efficiency	per cent.	84·5	78·6	81·6	77·5	78·0
Fuel per I.H.P. per hour	lb.	0·740	0·614	0·455	0·359	0·398
Fuel per B.H.P. per hour	lb.	0·876	0·781	0·558	0·463	0·510
Indicated thermal efficiency	per cent.	19·1	23·0	31·1	39·4	35·5
Brake thermal efficiency	per cent.	16·1	18·1	25·3	30·5	27·7

Fuel consumptions reduced to fuel having a calorific value of 10,000 C.H.U. (18,000 B.Th.U.) per lb.

* With crank case compression for scavenging and charging with air.

400. Comparative Trials of "Still" and "Sulzer" Engines.—Mr. William Denny, in a paper read before the Institution of Naval Architects in July, 1920, has described some interesting comparative trials of Still and Sulzer engines under actual working conditions on

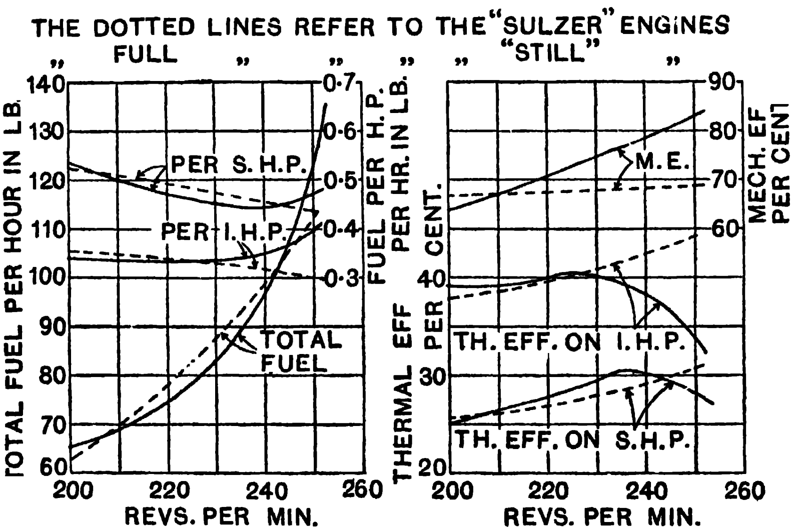


Fig. 654. Fig. 655.

board ship. The Still engine, the general features of which have been described in Art. 369, p. 512, is a combined internal combustion and steam engine. The Sulzer engine is of the two-stroke cycle Diesel type.

The Still engines tested had four cylinders per set, $7\frac{1}{4}$ inches in diameter and a piston stroke of 15 inches, and were of the opposed piston type, that is, each cylinder had two pistons moving outwards

and inwards from and to the centre of the cylinder. The steam boiler was of the Yarrow type. The Sulzer engines had also four cylinders per set, $8\frac{1}{2}$ inches in diameter and a piston stroke of $13\frac{3}{4}$ inches.

In each case the installation consisted of two sets, one for each propeller of the twin-screw ship in which the engines were tested. The conditions under which the trials were made were as nearly as possible identical, and the same oil fuel was used throughout. The fuel was shale oil having a specific gravity of 0·85 at 60° F. and a higher calorific value of about 19,000 B.Th.U.

The results of the trials are exhibited in Figs. 654 and 655. The falling away in the thermal efficiencies of the Still engines at the higher speeds will be noticed; this seems to indicate that the combustion of the oil fuel was incomplete at the higher speeds in these engines.

401. Use of Exhaust-Gas Charts.—From a complete analysis of the exhaust gases from an internal combustion engine, the ratio of the amount of air to fuel used may be found, also it can be seen whether the combustion is complete or not. In conjunction with an exhaust-gas chart for the fuel used, and a partial analysis of the exhaust gases,

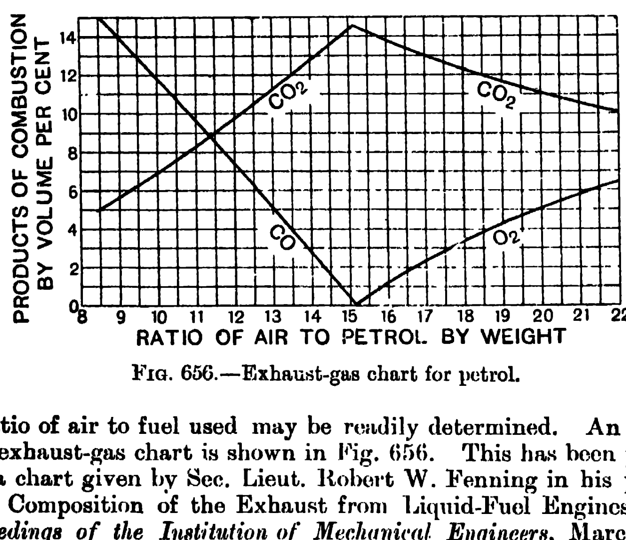


FIG. 656.—Exhaust-gas chart for petrol.

the ratio of air to fuel used may be readily determined. An example of an exhaust-gas chart is shown in Fig. 656. This has been prepared from a chart given by Sec. Lieut. Robert W. Fenning in his paper on "The Composition of the Exhaust from Liquid-Fuel Engines," in the *Proceedings of the Institution of Mechanical Engineers*, March, 1916. The particular chart shown in Fig. 656 is for petrol ("Bowley's Special"). Such a chart is prepared by plotting the results of analyses of the products of combustion of numerous mixtures of air and petrol. Examining the chart it will be seen that for a mixture of, say, 18 of air to one of petrol, by weight, the products of combustion contain, by volume, 12·3 per cent. of CO_2 and 3·4 per cent. of O_2, showing that there is an excess of air and that the combustion is complete. Again, for a mixture of, say, 10 of air to 1 of petrol the products of combustion contain 6·9 per cent. of CO_2 and 11·8 per cent. of CO, showing that the combustion is incomplete. It should be mentioned that the products of combustion referred to are the *dry* products, the water vapour being condensed and neglected. It will now be obvious that

from a partial analysis of the exhaust gases from an engine, and a reference to the exhaust-gas chart, the ratio of air to fuel used can be quickly determined. Of course each particular fuel has its own particular exhaust-gas chart.

402. The Tookey Factor.—The performances of all forms and sizes of internal combustion engines, whatever the kind of fuel used, may be compared by means of what is now known as the *Tookey factor.* This factor was originated by Mr. W. A. Tookey in his paper on "Commercial Tests of Internal Combustion Engines" in the *Proceedings of the Institution of Mechanical Engineers* for 1914.

Let P = mean effective pressure in lb. per square inch from the positive loop of the indicator diagram.
S = mixture strength in heat units per cubic foot of cylinder charge.
T = Tookey factor.

Then, $T = \dfrac{P}{S}$

The value of S is computed as follows.—

Let H = average calorific value of fuel per lb. or per cubic foot.
F = fuel used per minute in lb. or cubic feet.
I = number of impulses per minute.
V = effective or charge volume of cylinder in cubic feet.

Then, $S = \dfrac{H \times F}{I \times V} = \dfrac{\text{Heat supplied per impulse}}{\text{Effective cylinder volume}}$.

The effective cylinder volume is: stroke volume × volumetric efficiency + clearance volume.

According to Mr. Tookey, the value of the Tookey factor in the case of a modern gas engine at full load is about 2·3 when the heat is measured in B.Th.U., which corresponds to 4·14 when the heat is measured in C.H.U.

Exercises XXIV

1. The following data relate to a test of a four-stroke cycle gas engine. Duration of test, 50 minutes. Indicated horse-power, 37·4. Total gas used 452 cubic feet. Calorific value of the gas, 279 C.H.U. per cubic foot. Compression ratio 5·6. Determine the actual thermal efficiency and the relative efficiency.

2. A gas engine cylinder is 9 inches in diameter, piston stroke 16 inches, mean pressure from indicator diagram 75 lb. per square inch, revolutions 180 per minute, explosions 85 per minute, diameter of fly-wheel 5 feet, friction of brake-band 160 lb., gas used per hour 250 cubic feet of a thermal value of 600 B.Th.U. per cubic foot. Find the I.H.P., B.H.P., thermal efficiency, and mechanical efficiency of the engine. [Inst. C.E.]

3. A brake test of a gas engine gave the following results.—Speed 160 revolutions per minute. Diameter of brake-wheel 71 inches. Weight 310 lb., less mean spring-balance reading 37 lb. Gas consumed 402 cubic feet per hour. Calorific value of gas 670 B.Th.U. per cubic foot. Calculate the brake horse-power, the consumption of gas per brake horse-power-hour, and the thermal efficiency. [Inst. C.E.]

4. The indicated thermal efficiency of a gas engine is 33 per cent. The

calorific value of the gas used is 289 C.H.U., and the indicated horse-power is 38. Compute the gas consumption in cubic feet per hour.

5. An Otto cycle gas engine has a cylinder 10 inches diameter, and the stroke of the piston is 18 inches. Revolutions per minute, 180; miss-fires per minute, 8; mean effective pressure, 95 lb. per square inch; volume of gas used per hour, 430 cubic feet; calorific power of gas, 650 B.Th.U. per cubic foot. Determine—(*a*) the indicated horse-power; (*b*) the thermal efficiency; (*c*) the volume of gas consumed per indicated horse-power-hour. [U.L.]

6. The following particulars were obtained from a gas engine:—Cylinder 5·5 inches diameter, stroke 10 inches, clearance 22 per cent. of the volume displaced by the piston, I.H.P. 5·7, gas supplied per hour 87·5 cubic feet, calorific value of the gas 530 B.Th U. per cubic foot. Find the thermal efficiency of the engine and the relative efficiency when compared with the standard air engine (constant volume) having the same ratio of compression. [U.L.]

7. In a one hour's test of a gas engine 615 cubic feet of gas were used. The gas weighed 0·04 lb. per cubic foot and had a lower calorific value of 275 C.H.U. per cubic foot. The amount of air supplied was 6080 cubic feet and its density 0·08 lb. per cubic foot. The temperature of the exhaust gases near the exhaust valve was 512° C. and the temperature of the gas and air on entering the cylinder of the engine was 18° C. The exhaust gases contained 6·4 per cent., by weight, of steam. Taking the mean specific heat of the dry exhaust gases as 0·24 and the specific heat of the steam as 0·5, compute the heat of combustion carried away in the exhaust gases.

8. A four-stroke cycle gas engine has a cylinder 11·5 inches in diameter and a piston stroke of 21 inches. The compression ratio is 6·2. The effective diameter of the brake wheel is 71 inches. The observations made in a test of this engine were as follows:—Duration of test, 45 minutes. Total number of revolutions, 8145. Total number of explosions, 3465. Net load on brake, 314 lb. Mean effective pressure from indicator diagrams, 92 lb. per square inch. Gas used, 467 cubic feet. Pressure of gas at meter, 4·8 inches of water. Atmospheric temperature, 17° C. Height of barometer, 29·8 inches. Calorific value of gas, 275 C.H.U. per cubic foot at standard temperature (0° C.) and pressure (29·92 inches of mercury). Weight of water passing through the jacket, 693 lb. Temperatures of water at inlet and outlet of jacket, 17° C. and 68° C. respectively.

Find—(*a*) the I.H.P.; (*b*) the B.H.P.; (*c*) the mechanical efficiency; (*d*) the indicated thermal efficiency; and (*e*) the relative efficiency. Also draw up a percentage heat balance for the test.

9. The indicated horse-power of a gas engine is 9 at no load and 50 at full load. The mechanical efficiency at full load is 80 per cent. The speed being the same throughout, determine the probable mechanical efficiency at quarter load, half load, and three-quarter load.

10. In tests of a gas engine it was found that when the brake horse-power B was 22·31, the gas consumption G was 410 cubic feet per hour. At the same speed when B was 13·7, G was 298. Assuming that the relation between G and B is a linear one, find this relation and use it to calculate G when B is 10·6.

11. If for a gas engine the total gas consumption per hour at no load is denoted by M, and at full load by M + N, show that the equation to the Willans line is, $G = M + xN$, where G is the gas consumption per hour at the fraction x of full load.

12. An aero engine uses 0·5 lb. of petrol per B.H.P. per hour. The calorific value of the petrol is 10,400 C.H.U. per lb. Assuming that 30 per cent. of the heat supplied is carried away by the jacket water and that the rise in temperature of the water as it passes through the jacket is 25° C., what weight of water must pass through the jacket per B.H.P. per minute?

The heated water is returned to its original temperature by passing it through a radiator. Determine the minimum volume of air which must pass through the radiator per B.H.P. per minute, assuming that its specific heat is 0·24, and that 1 lb. of air has a volume of 13 2 cubic feet.

13. A certain gas has a calorific value of 270 C.H.U. per cubic foot. A mixture of this gas and air contains 0·18 cubic foot of gas and 1·53 cubic feet of air. What is the calorific value of 1 cubic foot of the mixture?

14. If the volumetric efficiency of a petrol engine is 80 per cent. and the mixture in the cylinder before compression has a calorific value of 54 C.H.U. per

cubic foot, and if the thermal efficiency is 31 per cent., compute the mean effective pressure in the cylinder.

15. A published report of certain tests of a four-cylinder petrol engine for a motor car gave curves showing the petrol consumption per brake horse-power per hour from which the figures in the following table have been taken.

Consumption of Petrol in lb. per B.H.P. per Hour.

Revs. per min.	Brake Horse-power.					30	35		
1450	--		0·967	0·816	--	0·667	—	0·624	0·611
1050	1·232	1·036	0·832	0·722	0·657	—	0·608		
625	0·906	0·830	0·720	0·643	0·612	—	—		

Compute the total consumption of petrol in lb. per hour for each of the results in the table, and then plot them on a B.H.P. base and draw the Willans line for each speed and find its equation.

Lastly, taking the calorific value of the petrol as 10,330 C.H.U. per lb., construct the brake thermal efficiency curves on a B.H.P. base for each of the three speeds given.

16. The following data relate to a 180-H.P. Mercedes aero engine tested at the English Royal Aircraft Factory in 1918. Number of cylinders, 6; bore, 140 mm. (5·51 in.); stroke, 160 mm. (6·30 in.); compression ratio, 4·64; B.H.P., 174, developed at the normal speed of 1400 revs. per min.; consumption of petrol *in pints* per B.H.P. per hour, 0·545.

Calculate the M.E.P. equivalent of the B.H.P. Assuming that the specific gravity of the petrol was 0·72 and that its lower calorific value was 10,400 C.H.U. per lb., calculate the brake thermal efficiency. Also assuming a mechanical efficiency of 82 per cent. compute the actual M.E.P., the indicated thermal efficiency, and the relative efficiency.

17. A motor car engine has four cylinders 80 mm. (3·15 in.) in diameter and a piston stroke of 13 mm. (5·12 in.). In a test of this engine lasting 30 minutes the following observations were made.—Average speed, 1204 revolutions per minute; petrol consumption, 7·04 lb.; calorific value of petrol, 10,400 C.H.U. per lb. The brake horse-power was measured with a fan brake for which the formula was B.H.P. $= \frac{13N^3}{10^9}$ where N is the number of revolutions per minute.

Calculate—(*a*) The B.H.P.; (*b*) the petrol consumption per B.H.P. per hour; (*c*) the brake thermal efficiency. Also, (*d*) assuming that the mechanical efficiency of the engine was 85 per cent., what was the M.E.P. in the cylinders?

18. The following particulars were obtained from trials of a four-stroke cycle oil engine:—cylinder diam., 12 in.; stroke, 18·25 in.; diam. of brake wheel, 9 ft. 0·75 in.

Average revs. per min.	199	202	203	204	204	205
Brake load . . . . lb.	225	195	156	113	60	0
Mean effective pressure lb. per sq in.	107	97	87	74	60	42
Oil per hour . . . . . . lb.	20·7	16·3	13·5	10·7	10·0	9·5

Draw to a base of B.H.P. curves showing mechanical efficiency and oil used per hour per B.H.P. [U.L.]

19. On the diagram in answer to the preceding exercise draw the curve showing oil used per hour per I.H.P.

20. Taking the data of exercise 18 and the lower calorific value of the oil as

10,000 C.H.U. (18,000 B.Th.U.) per lb. draw, on a B.H.P. base, the curves showing oil per hour, indicated thermal efficiency, and brake thermal efficiency.

21. The normal full load on a certain two-stroke cycle marine Diesel engine is 600 B.H.P. Tests of this engine gave, (*a*) 733 I.H.P. with a load of 566 B.H.P., and (*b*) 480 I.H.P. with a load of 316 B.H.P. In both tests the speed was very approximately 200 revolutions per minute. Assuming that the relation between the I.H.P. and the B.H.P. is a linear one, deduce a formula for the mechanical efficiency of this engine in terms of x the ratio of the load to full load. Apply this formula to calculate the mechanical efficiency at quarter load, half load, three-quarter load, full load, and ten per cent. overload.

The tests (*a*) and (*b*) gave indicated thermal efficiencies of 40·5 per cent. and 43·8 per cent. respectively. Assuming that the indicated thermal efficiencies plot into a straight line on a load base, which was proved to be very approximately the case in quite a number of tests of this engine, find the equation to this line in terms of x. Compute the indicated thermal efficiency and the brake thermal efficiency at the five loads mentioned above. For what value of x is the brake thermal efficiency a maximum? Taking the calorific value of the fuel as 10,000 C.H.U. (18,000 B.Th.U.) per lb. compute the fuel consumption in lb. per I.H.P. and per B.H.P. per hour at the same five loads. Lastly, plot all the results on a load base.

22. At full load the indicated horse-power of a Diesel engine was 690. The indicated horse-power of the compressor was 39·9, and the fuel oil used was 224 lb. per hour. The lower calorific value of the fuel was 10,450 C.H.U. (18,810 B.Th.U.) per lb. Calculate the fuel consumption per hour per diagram I.H.P. and per net I.H.P. Compute also the corresponding thermal efficiencies.

23. The fuel consumption of a Diesel engine, in lb. per B.H.P. per hour, was found to be, 0·636, 0·446, 0·409, 0·411, and 0·430, at quarter load, half load, three-quarter load, full load, and 20 per cent. overload respectively. The calorific value of the fuel was 10,850 C.H.U. (19,530 B.Th.U.) per lb. Calculate the corresponding consumptions of a fuel having a calorific value of 10,000 C.H.U. (18,000 B.Th.U.) per lb. Plot both sets of results.

24. The analysis by weight of a sample of crude petroleum used in an oil engine gives, carbon 85 per cent., hydrogen 13·5 per cent., and incombustible matter 1·5 per cent. The volume analysis of the exhaust gases gives, CO_2 7 per cent., O 11·3 per cent., and N 81·7 per cent. The engine uses 0·333 lb. of oil per I.H.P. per hour, and 14·8 lb. of water per I.H.P. per hour pass through the jacket. The rise in temperature of the jacket water is 52° C. (93·6° F.). The temperature of the exhaust is 384° C. (723·2° F.). The lower calorific value of the oil is 10,720 C.H.U. (19,296 B.Th.U.) per lb., and the specific heat of the exhaust gases is 0 24. From the above data draw up a heat balance for the engine. [U.L.]

APPENDIX

Densities and Volumes of Gases.—Table I gives the specific densities and specific volumes of various gases.

W = weight in lb. of 1 cubic foot of gas.

V = volume in cubic feet of 1 lb. weight of gas. V = 1/W.

In each case the temperature of the gas is 0° C., and the pressure 760 *mm.* of mercury, or 14·7 lb. per square inch.

TABLE I.—*Specific Densities and Specific Volumes of Gases.*

Gas.	W	V
Hydrogen	0·0056	178·57
Oxygen	0·0892	11·21
Nitrogen	0·0781	12·80
Air	0·0807	12·39
Carbon dioxide, CO_2	0·1234	8·10
Carbon monoxide, CO	0·0781	12·80
Methane (Marsh gas), CH_4	0·0448	22·32
Ethylene (Olefiant gas), C_2H_4	0·0795	12·58
Acetylene, C_2H_2	0·0743	13·46
Sulphur dioxide, SO_2	0·1827	5·47

If v is the volume of a given quantity of a gas at a temperature $t°$ C., and pressure h mm. of mercury, or p lb. per square inch, and if v_0 is the volume of the same quantity of the gas at 0° C. and 760 mm. of mercury, or 14·7 lb. per square inch, then

$$v_0 = \frac{v}{1 + 0{\cdot}00367t} \cdot \frac{h}{760} = \frac{v}{1 + 0{\cdot}00367t} \cdot \frac{p}{14{\cdot}7}$$

Steam Tables.—Tables II, III, and IV give the properties of saturated steam.

The columns to the left of the first thick line and the columns between the thick lines apply to degrees centigrade and C.H.U.

The columns to the right of the second thick line and the columns between the thick lines apply to degrees Fahrenheit and B.Th.U.

t = saturation temperature.

H = total heat of 1 lb. of steam.

L = latent heat of 1 lb. of steam.

h (not given) = H − L = sensible heat of 1 lb. of steam, or the heat in 1 lb. of water at temperature t.

p = pressure of steam in lb. per square inch, absolute.

v = volume of 1 lb. of steam, in cubic feet.

w (not given) = weight of 1 cubic foot of steam = $1/v$, in lb.

ϕ_w = entropy of 1 lb. of water.

ϕ_s = total entropy of 1 lb. of steam.

ϕ_e (not given) = entropy of evaporation of 1 lb. of steam = $\phi_s - \phi_w$.

Tables VI, VII, VIII, and IX give the properties of superheated steam.

In Table III the leading column is *temperature*; in the other steam tables the leading column is *pressure*.

Specific Volume of Water.—Table V gives the specific volume u' of water at various temperatures. The specific density u, or the weight in lb. of 1 cubic foot of water = $1/u'$.

TABLE II.—*Properties of Saturated Steam.*

$t°$ C.	H C.H.U.	L C.H.U.	p	v	ϕ_w	ϕ_s	L B.Th.U.	H B.Th.U.	$t°$ F
38·78	613·0	574·3	1	333·1	0·1325	1·9738	1033·7	1103·4	101·8
52·28	619·1	566·9	2	173·5	0·1748	1·9170	1020·4	1114·3	126·1
60·83	622·8	562·1	3	118·5	0·2007	1·8839	1011·8	1121·1	141·5
67·22	625·6	558·5	4	90·5	0·2197	1·8607	1005·3	1126·1	153·0
72·39	627·8	555·6	5	73·4	0·2347	1·8427	1000·0	1130·1	162·3
76·72	629·7	553·1	6	61·90	0·2471	1·8280	995·5	1133·4	170·1
80·50	631·3	550·9	7	53·55	0·2579	1·8158	991·6	1136·3	176·9
83·83	632·7	548·9	8	47·28	0·2673	1·8050	988·0	1138·8	182·9
86·83	633·9	547·1	9	42·36	0·2757	1·7957	984·8	1141·0	188·3
89·57	635·1	545·6	10	38·38	0·2833	1·7874	982·0	1143·1	193·2
92·11	636·1	544·0	11	35·11	0·2903	1·7798	979·2	1144·9	197·8
94·44	637·0	542·6	12	32·38	0·2967	1·7729	976·7	1146·6	202·0
96·61	637·8	541·3	13	30·04	0·3026	1·7666	974·3	1148·1	205·9
98·67	638·7	540·0	14	28·02	0·3081	1·7607	972·0	1149·6	209·6
100·0	639·2	539·2	14·7	26·79	0·3118	1·7570	970·6	1150·6	212·0
100·6	639·4	538·8	15	26·27	0·3133	1·7553	969·9	1150·9	213·0
102·4	640·1	537·7	16	24·74	0·3183	1·7502	967·8	1152·2	216·3
104·1	640·8	536·6	17	23·38	0·3229	1·7454	965·9	1153·4	219·4
105·8	641·4	535·6	18	22·16	0·3273	1·7408	964·0	1154·5	222·4
107·3	642·0	534·6	19	21·07	0·3315	1·7365	962·3	1155·6	225·2
108·9	642·6	533·6	20	20·08	0·3355	1·7324	960·5	1156·6	228·0
110·3	643·1	532·7	21	19·18	0·3393	1·7285	958·8	1157·6	230·6
111·7	643·6	531·8	22	18·37	0·3430	1·7248	957·2	1158·5	233·1
113·1	644·1	530·9	23	17·62	0·3465	1·7213	955·6	1159·4	235·5
114·3	644·6	530·1	24	16·93	0·3499	1·7179	954·1	1160·2	237·8
115·6	645·0	529·2	25	16·29	0·3532	1·7146	952·6	1161·0	240·1
116·8	645·4	528·4	26	15·71	0·3564	1·7116	951·2	1161·8	242·3
118·0	645·9	527·7	27	15·17	0·3594	1·7087	949·9	1162·6	244·4
119·1	646·3	527·0	28	14·67	0·3623	1·7059	948·6	1163·4	246·4
120·2	646·7	526·3	29	14·19	0·3652	1·7032	947·3	1164·1	248·4
121·3	647·1	525·6	30	13·74	0·3680	1·7006	946·0	1164·8	250·3
123·3	647·8	524·2	32	12·93	0·3733	1·6954	943·5	1166·0	254·0
125·3	648·4	522·8	34	12·22	0·3783	1·6904	941·0	1167·2	257·6
127·2	649·1	521·6	36	11·58	0·3831	1·6859	938·8	1168·4	261·0
129·0	649·7	520·3	38	11·01	0·3876	1·6816	936·6	1169·5	264·2
130·7	650·3	519·2	40	10·49	0·3919	1·6776	934·6	1170·6	267·3
132·3	650·9	518·1	42	10·02	0·3961	1·6739	932·6	1171·6	270·2
133·9	651·4	517·0	44	9·59	0·4001	1·6702	930·6	1172·5	273·0
135·4	651·9	515·9	46	9·20	0·4039	1·6667	928·7	1173·4	275·7
136·9	652·3	514·9	48	8·84	0·4075	1·6633	926·8	1174·2	278·4
138·3	652·8	513·8	50	8·51	0·4111	1·6600	924·9	1175·0	281·0
139·7	653·2	512·9	52	8·20	0·4145	1·6569	923·2	1175·8	283·5
141·1	653·7	511·9	54	7·92	0·4178	1·6540	921·5	1176·6	285·9
142·3	654·1	511·0	56	7·65	0·4210	1·6512	919·8	1177·3	288·2
143·6	654·4	510·2	58	7·40	0·4241	1·6485	918·3	1178·0	290·4

TABLE II. (*continued*)—*Properties of Saturated Steam.*

$t°$ C.	H C.H.U.	L C.H.U.	p	v	ϕ_w	ϕ_s	L B.Th.U.	H B.Th.U.	$t°$ F.
144·8	654·8	509·3	60	7·17	0·4271	1·6458	916·7	1178·7	292·6
146·0	655·2	508·4	62	6·95	0·4300	1·6431	915·2	1179·4	294·8
147·2	655·6	507·6	64	6·75	0·4328	1·6405	913·6	1180·0	296·9
148·3	655·9	506·8	66	6·56	0·4356	1·6381	912·2	1180·6	298·9
149·4	656·2	505·9	68	6·38	0·4383	1·6358	910·7	1181·2	300·9
150·5	656·6	505·2	70	6·20	0·4409	1·6335	909·3	1181·8	302·9
151·6	656·9	504·4	72	6·04	0·4435	1·6312	907·9	1182·4	304·8
152·6	657·2	503·6	74	5·89	0·4460	1·6290	906·5	1182·9	306·7
153·6	657·4	502·8	76	5·74	0·4485	1·6269	905·1	1183·4	308·5
154·6	657·7	502·1	78	5·60	0·4509	1·6249	903·8	1183·9	310·3
155·6	658·0	501·4	80	5·47	0·4533	1·6229	902·5	1184·4	312·0
156·5	658·3	500·7	82	5·35	0·4555	1·6209	901·2	1184·9	313·7
157·4	658·6	499·9	84	5·23	0·4577	1·6190	899·9	1185·4	315·4
158·3	658·8	499·3	86	5·11	0·4599	1·6172	898·8	1185·9	317·0
159·2	659·1	498·7	88	5·00	0·4621	1·6154	897·6	1186·4	318·6
160·1	659·3	498·0	90	4·90	0·4642	1·6136	896·4	1186·8	320·2
161·0	659·6	497·3	92	4·80	0·4662	1·6118	895·1	1187·2	321·8
161·8	659·8	496·6	94	4·70	0·4682	1·6101	893·9	1187·6	323·3
162·7	660·0	496·0	96	4·61	0·4702	1·6084	892·8	1188·0	324·8
163·5	660·2	495·3	98	4·52	0·4722	1·6067	891·6	1188·4	326·3
164·3	660·4	494·7	100	4·44	0·4741	1·6050	890·5	1188·8	327·8
166·3	660·9	493·2	105	4·24	0·4787	1·6011	887·7	1189·7	331·3
168·2	661·4	491·7	110	4·06	0·4832	1·5974	885·1	1190·6	334·7
170·0	661·9	490·3	115	3·89	0·4875	1·5939	882·5	1191·5	338·0
171·8	662·4	488·9	120	3·73	0·4917	1·5906	880·0	1192·3	341·2
173·5	662·8	487·6	125	3·59	0·4957	1·5874	877·6	1193·1	344·3
175·2	663·3	486·3	130	3·46	0·4996	1·5843	875·3	1193·9	347·3
176·8	663·7	484·9	135	3·34	0·5033	1·5813	872·9	1194·6	350·2
178·3	664·1	483·7	140	3·23	0·5070	1·5784	870·7	1195·3	353·0
179·8	664·4	482·4	145	3·12	0·5105	1·5756	868·4	1195·9	355·7
181·3	664·7	481·2	150	3·02	0·5140	1·5729	866·2	1196·5	358·4
182·8	665·1	480·1	155	2·93	0·5173	1·5703	864·1	1197·1	361·0
184·2	665·4	478·9	160	2·84	0·5206	1·5678	862·0	1197·7	363·5
185·6	665·7	477·7	165	2·76	0·5238	1·5653	859·9	1198·3	366·0
186·9	666·0	476·6	170	2·68	0·5268	1·5629	857·9	1198·8	368·4
189·5	666·6	474·5	180	2·54	0·5327	1·5583	854·1	1199·9	373·1
192·0	667·2	472·4	190	2·42	0·5383	1·5539	850·3	1200·9	377·6
194·4	667·7	470·3	200	2·30	0·5437	1·5498	846·6	1201·8	381·9
196·7	668·1	468·4	210	2·20	0·5489	1·5459	843·1	1202·6	386·0
198·9	668·6	466·4	220	2·10	0·5539	1·5421	839·6	1203·4	390·0
205·1	669·8	461·0	250	1·86	0·5678	1·5319	829·8	1205·6	401·1
214·2	671·4	452·8	300	1·56	0·5882	1·5173	815·0	1208·6	417·6
222·3	672·8	445·4	350	1·35	0·6060	1·5051	801·7	1211·0	432·1
229·5	674·0	438·7	400	1·19	0·6217	1·4945	789·6	1213·2	445·1
236·1	674·8	432·1	450	1·06	0·6362	1·4848	777·8	1214·7	457·0

TABLE III.—*Properties of Saturated Steam.*

$t°$ C.	H C.H.U.	L C.H.U.	p	v	ϕ_w	ϕ_s	L B.Th.U.	H B.Th.U.	$t°$ F.
10	599·8	589·8	0·178	1700	0·0360	2·1194	1061·7	1079·7	50
15	602·2	587·2	0·248	1246	0·0535	2·0915	1056·9	1083·9	59
20	604·5	584·5	0·339	925	0·0706	2·0648	1052·1	1088·1	68
25	606·8	581·8	0·459	695	0·0875	2·0392	1047·3	1092·2	77
30	609·1	579·1	0·614	528	0·1040	2·0146	1042·4	1096·3	86
35	611·3	576·4	0·814	404·4	0·1203	1·9910	1037·5	1100·4	95
40	613·6	573·6	1·068	313·1	0·1364	1·9684	1032·5	1104·4	104
45	615·8	570·8	1·387	244·5	0·1522	1·9468	1027·5	1108·4	113
50	618·0	568·1	1·786	192·7	0·1678	1·9261	1022·6	1112·4	122
55	620·2	565·3	2·280	153·2	0·1832	1·9062	1017·6	1116·4	131
60	622·4	562·6	2·89	122·8	0·1983	1·8871	1012·6	1120·4	140
65	624·7	559·8	3·62	99·2	0·2131	1·8687	1007·6	1124·4	149
70	626·8	556·9	4·52	80·7	0·2278	1·8510	1002·5	1128·3	158
75	629·0	554·1	5·59	66·2	0·2422	1·8339	997·4	1132·2	167
80	631·1	551·2	6·87	54·5	0·2565	1·8174	992·2	1136·0	176
85	633·2	548·3	8·38	45·26	0·2706	1·8015	986·9	1139·7	185
90	635·2	545·3	10·17	37·79	0·2845	1·7861	981·5	1143·4	194
95	637·2	542·3	12·25	31·74	0·2982	1·7712	976·1	1147·0	203
100	639·2	539·2	14·70	26·79	0·3118	1·7568	970·6	1150·6	212
105	641·2	536·1	17·52	22·73	0·3252	1·7429	965·0	1154·1	221
110	643·0	532·9	20·8	19·38	0·3385	1·7295	959·2	1157·4	230
115	644·8	529·7	24·5	16·59	0·3516	1·7165	953·4	1160·7	239
120	646·6	526·4	28·8	14·28	0·3646	1 7038	947·5	1163·9	248
125	648·4	523·1	33·7	12·33	0·3775	1·6914	941·5	1167·1	257
130	650·1	519·7	39·2	10·70	0·3902	1·6793	935·4	1170·1	266
135	651·7	516·2	45·4	9·31	0·4028	1·6676	929·2	1173·1	275
140	653·3	512·7	52·4	8·14	0·4153	1·6562	922·9	1176·0	284
145	654·9	509·1	60·3	7·14	0·4275	1·6451	916·4	1178·8	293
150	656·4	505·5	69·1	6·28	0·4397	1·6343	909·9	1181·5	302
155	657·8	501·8	78·9	5·55	0·4519	1·6238	903·2	1184·1	311
160	659·3	498·1	89·8	4·92	0·4639	1·6136	896·5	1186·7	320
165	660·6	494·2	101·8	4·37	0·4757	1·6036	889·5	1189·1	329
170	661·9	490·3	115·0	3·89	0·4875	1·5939	882·5	1191·5	338
175	663·2	486·3	129·5	3·47	0·4992	1·5845	875·4	1193·8	347
180	664·4	482·3	145·5	3·11	0·5109	1·5754	868·2	1196·0	356
185	665·6	478·2	163·0	2·80	0·5224	1·5664	860·8	1198·1	365
190	666·7	474·1	182·0	2·52	0·5339	1·5575	853·3	1200·1	374
195	667·8	469·8	202·7	2·27	0·5452	1·5488	845·6	1202·0	383
200	668·8	465·4	225·2	2·05	0·5565	1·5403	837·8	1203·8	392
205	669·7	461·1	249·6	1·86	0·5677	1·5320	829·9	1205·5	401
210	670·7	456·6	276·0	1·69	0·5788	1·5239	821·9	1207·2	410
215	671·6	452·1	304·5	1·53	0·5899	1·5160	813·8	1208·8	419
220	672·4	447·5	335·2	1·40	0·6009	1·5083	805·5	1210·3	428
225	673·2	442·9	368·2	1·29	0·6119	1·5008	797·2	1211·8	437
230	674·1	438·2	403·6	1·18	0·6228	1·4935	788·7	1213·3	446

TABLE IV.—*Estimated Properties of Saturated Steam at very High Pressures.*[1]

$t°$ C.	C.H.U.	H C.H.U.	L C.H.U.	p	v	L B.Th.U.	H B.Th.U.	h B.Th.U.	$t°$ F.
315·6	335·6	646·7	311·1	1540	0·272	560	1164	604	600
326·7	351·7	639·4	287·8	1659	0·241	518	1151	633	620
337·8	368·3	631·1	262·8	2057	0·187	473	1136	663	640
348·9	388·9	617·8	228·9	2361	0·151	412	1112	700	660
360·0	413·9	600·0	186·1	2699	0·118	335	1080	745	680
371·1			107·2	3075	0·080	193	1016	823	700
374·6*	511·7	511·7	0·0	3200	0·048	0	921	921	706·3*

* Critical temperature (see pp. 11 and 56).

[1] "Thermal Properties of Steam," by G. A. Goodenough, Bulletin No. 75, University of Illinois, 1914.

TABLE V.—*Specific Volume of Water.*

Specific volume $= u' =$ volume of 1 lb. in cubic feet.

$t°$ C.	$t°$ F.	u'	$t°$ C.	$t°$ F.	u'	$t°$ C.	$t°$ F.	u'
0	32	0·0160	100	212	0·0167	200	392	0·0186
10	50	0·0160	110	230	0·0168	210	410	0·0189
20	68	0·0160	120	248	0·0169	220	428	0·0192
30	86	0·0161	130	266	0·0171	230	446	0·0195
40	104	0·0161	140	284	0·0173	240	464	0·0198
50	122	0·0162	150	302	0·0175	250	482	0·0202
60	140	0·0163	160	320	0·0177	260	500	0·0206
70	158	0·0164	170	338	0·0179	270	518	0·0210
80	176	0·0165	180	356	0·0181	280	536	0·0215
90	194	0·0166	190	374	0·0183	290	554	0·0221

TABLE VI.—*Total Heat of Superheated Steam in C.H.U.*

p = pressure in lb. per sq. in. abs. t = temperature of saturation

p	$t°$ C.	Degrees of Superheat (Centigrade). 0	20	40	60	80	100	150	200
20	108·9	642·6	652·4	662·1	671·8	681·4	691·0	714·9	738·7
30	121·3	647·1	657·1	666·9	676·7	686·4	696·1	720·2	744·1
40	130·7	650·3	660·4	670·4	680·3	690·2	700·0	724·2	748·3
50	138·3	652·8	663·1	673·2	683·3	693·2	703·1	727·4	751·6
60	144·8	654·8	665·3	675·6	685·7	695·7	705·6	730·1	754·3
70	150·5	656·6	667·2	677·6	687·7	697·8	707·7	732·3	756·6
80	155·6	658·0	668·8	679·3	689·6	699·7	709·7	734·3	758·7
90	160·1	659·3	670·2	680·8	691·2	701·4	711·4	736·1	760·6
100	164·3	660·4	671·5	682·2	692·7	702·9	712·9	737·8	762·3
120	171·8	662·4	673·7	684·6	695·2	705·4	715·6	740·6	765·2
140	178·3	664·1	675·7	686·7	697·4	707·8	718·0	743·0	767·7
160	184·2	665·4	677·3	688·5	699·2	709·7	720·0	745·2	769·9
180	189·5	666·6	678·8	690·1	700·9	711·4	721·8	747·1	771·9
200	194·4	667·7	680·1	691·6	702·4	713·0	723·4	748·8	773·8
250	205·1	669·8	682·8	694·6	705·6	716·3	726·8	752·5	777·6
300	214·2	671·4	685·2	697·2	708·4	719·2	729·8	755·8	781·1

TABLE VII.—*Total Heat of Superheated Steam in B Th.U.*

p = pressure in lb. per sq in abs. t = temperature of saturation.

p	$t°$ F.	Degrees of Superheat (Fahrenheit). 0	36	72	108	144	180	270	360
20	228·0	1156·6	1174·3	1191·8	1209·2	1226·5	1243·8	1286·8	1329·7
30	250·3	1164·8	1182·7	1200·4	1218·0	1235·5	1253·0	1296·3	1339·4
40	267·3	1170·6	1188·8	1206·8	1224·6	1242·3	1260·0	1303·6	1346·9
50	281·0	1175·0	1193·5	1211·8	1229·9	1247·8	1265·5	1309·3	1352·8
60	292·6	1178·7	1197·5	1216·0	1234·3	1252·3	1270·1	1314·1	1357·8
70	302·9	1181·8	1200·9	1219·6	1237·9	1256·0	1273·9	1318·1	1361·9
80	312·0	1184·4	1203·8	1222·7	1241·2	1259·4	1277·4	1321·7	1365·6
90	320·2	1186·8	1206·4	1225·5	1244·2	1262·5	1280·5	1325·0	1369·0
100	327·8	1188·8	1208·7	1228·0	1246·8	1265·2	1283·3	1328·0	1372·1
120	341·2	1192·3	1212·7	1232·3	1251·3	1269·8	1288·1	1333·0	1377·3
140	353·0	1195·3	1216·2	1236·1	1255·3	1274·0	1292·4	1337·4	1381·8
160	363·5	1197·7	1219·1	1239·3	1258·6	1277·4	1296·0	1341·3	1385·9
180	373·1	1199·9	1221·8	1242·2	1261·6	1280·5	1299·2	1344·8	1389·5
200	381·9	1201·8	1224·1	1244·8	1264·4	1283·4	1302·1	1347·9	1392·8
250	401·1	1205·6	1229·1	1250·2	1270·1	1289·4	1308·3	1354·5	1399·7
300	417·6	1208·6	1233·4	1255·0	1275·2	1294·6	1313·7	1360·4	1406·0

Table VIII.—*Specific Volume of Superheated Steam.*

Volume in cubic feet per pound.

sur. sq.	Temp. of Saturation.	° F.		Degrees of Superheat (Centigrade). 20	40	60	80	100		
					Degrees of Superheat (Fahrenheit). 72	108	144	180	270	360
20	108·9	228·0	20·08	21·23	22·36	23·47	24·57	25·67	28·41	31·13
30	121·3	250·3	13·74	14·53	15·29	16·04	16·78	17·52	19·36	21·18
40	130·7	267·3	10·49	11·09	11·68	12·25	12·81	13·37	14·76	16·13
50	138·3	281·0	8·51	9·00	9·47	9·93	10·39	10·84	11·96	13·06
60	144·8	292·6	7·17	7·58	7·98	8·37	8·76	9·14	10·08	11·00
70	150·5	302·9	6·20	6·56	6·91	7·25	7·58	7·91	8·72	9·51
80	155·6	312·0	5·47	5·79	6·10	6·39	6·68	6·97	7·69	8·39
90	160·1	320·2	4·90	5·19	5·46	5·72	5·98	6·24	6·88	7·50
100	164·3	327·8	4·44	4·70	4·95	5·19	5·43	5·66	6·23	6·79
120	171·8	341·2	3·73	3·96	4·17	4·37	4·57	4·77	5·25	5·72
140	178·3	353·0	3·23	3·42	3·61	3·79	3·96	4·13	4·55	4·96
160	184·2	363·5	2·84	3·01	3·18	3·34	3·50	3·65	4·02	4·38
180	189·5	373·1	2·54	2·70	2·85	2·99	3·13	3·27	3·60	3·92
200	194·4	381·9	2·30	2·44	2·58	2·71	2·84	2·96	3·26	3·55
250	205·1	401·1	1·86	1·98	2·09	2·20	2·30	2·40	2·65	2·89
300	214·2	417·6	1·56	1·66	1·76	1·85	1·94	2·03	2·24	2·44

Table IX.—*Total Entropy of Superheated Steam.*

Pressu b. per sq	Temp. of Saturation			Degrees of Superheat (Centigrade). 40	60	80	100			
				Degrees of Superheat (Fahrenheit). 72	108	144	180		270	360
20	108·9	228·0	1·7324	1·7574	1·7810	1·8034	1·8247	1·8451	1·8922	1·9350
30	121·3	250·3	1·7006	1·7252	1·7484	1·7704	1·7913	1·8114	1·8577	1·8998
40	130·7	267·3	1·6776	1·7020	1·7251	1·7469	1·7677	1·7876	1·8334	1·8750
50	138·3	281·0	1·6600	1·6844	1·7074	1·7291	1·7498	1·7695	1·8149	1·8561
60	144·8	292·6	1·6458	1·6702	1·6931	·7147	1·7352	1·7547	1·7997	1·8406
70	150·5	302·9	1·6335	1·6580	1·6809	·7024	1·7228	1·7422	1·7870	1·8276
80	155·6	312·0	1·6229	1·6475	1·6704	·6919	1·7122	1·7315	1·7760	1·8164
90	160·1	320·2	1·6136	1·6382	1·6611	·6826	1·7028	1·7220	1·7663	1·8065
100	164·3	327·8	·6050	1·6297	1·6526	·6740	1·6941	1·7132	1·7573	1·7973
120	171·8	341·2	·5906	1·6155	1·6384	·6597	1·6797	1·6987	1·7425	1·7822
140	178·3	353·0	·5784	1·6035	1·6265	·6478	1·6677	1·6866	1·7301	1·7695
160	184·2	363·5	·5678	1·5932	1·6162	·6374	1·6572	1·6760	1·7193	1·7585
180	189·5	373·1	·5583	1·5840	1·6070	·6281	1·6478	1·6665	1·7096	1·7485
200	194·4	381·9	·5498	1·5758	1·5989	·6199	1·6395	1·6581	1·7010	1·7398
250	205·1	401·1	·5319	1·5586	1·5817	·6026	1·6221	1·6406	1·6832	1·7217
300	214·2	417·6	·5173	1·5450	1·5682	·5891	1·6085	1·6269	1·6692	1·7074

NUMERICAL ANSWERS TO EXERCISES

Exercises I, pp. 21, 22.

1. 982·2; 315·6; 95; 22·2; − 3·9; − 17·8; − 21·1.
2. 2732; 1022; 215·6; 68; 46·4; 3·2; − 4.
3. − 40. **4.** 0·081 inch. **5.** + 0·00972 inch. **6.** 35·863 inches.
7. 1·304 lb. **8.** 1194 C.H.U.; 0·0965. **9.** 0·0296.
10. $a = 0·111$; $b = 0·00004$. **11.** 80·4 C.H.U. **12.** 243·8 C.H.U.
13. 27·19 lb. **14.** 1·666 lb. **15.** 61·2° C. **16.** 72,320 B.Th.U. **17.** 0·6 per cent.

18. Q = 6114 C.H.U. (11,005 B.Th.U.). Temperature heads, per cent.—Gas film, 96·6; plate, 1·2; water film, 2·3. $a = 70·17$ for °C. and C.H.U., or 126·3 for °F. and B.Th.U.

19. Case I: Q = 8736 B.Th.U.; $a = 164·8$. Case II: Q = 14,820 B.Th.U.; $a = 97·2$.

20. (*a*) 2,688,000; (*b*) 2,484,472; (*c*) 6,732,000; (*d*) 6,635,389; (*e*) 11,718,000.

Exercises II, pp. 48–51.

1. 80 lb. per sq. inch. **2.** 55·62 lb. per sq. inch.
3. 13·974 cubic feet; 17·566 cubic feet; 3·069 lb.
4. 73·5 lb. per sq. inch; 5·696 cubic feet. **5.** 1384.
6. 802°C.; 143·3 lb. per sq. inch. **7.** 60·6°C.; 1·28. **8.** 20 cubic feet.

9. (*a*) 29·4 lb. per sq. inch; 13·44 cubic feet; 128·52 B.Th.U.; 91·26 B.Th.U.; 28,987 ft.-lb. or 37·26 B.Th.U. (*b*) 59·94 lb. per sq. inch; 6·59 cubic feet; 91·26 B.Th.U.; 91·26 B.Th.U.; 0.

10. 7016 ft.-lb. **11.** 46·51 lb. per sq. inch; 282·4°C.; 10,500 ft.-lb.
12. (*a*) 593·9°C. (1101° F.); 121,464 ft.-lb. (*b*) 777° C. (1430·6° F.); 155,040 ft.-lb.
13. 1·23. **14.** 1·27. **15.** $k_v = 0·168$; $k_p = 0·237$.
16. 57,024 ft.-lb.; 15·27 C.H.U. (27·49 B.Th.U.).
17. 3·346 C.H.U. (6·023 B.Th.U.). **18.** 142·6. **19.** 342,940 ft.-lb.
20. 26 inches. **21.** $C_3 = (C_2^{\frac{1}{n}} - C_1^{\frac{1}{n}})^n$; $C_4 = C_1^{\frac{1}{n}} \div (C_2^{\frac{1}{n}} - C_1^{\frac{1}{n}})$.
22. 989; 93. **23.** 101·4; 36·8 lb. per sq. inch. **24.** 75 per cent.
25. (*a*) 21·8 inches; (*b*) 25·6 inches.
26. (*a*) 26 81 lb.; (*b*) 32·54; (*c*) 49·43 lb.; (*d*) − 76·1° C. (− 105° F.).
27. 125·7° C. (258·3° F.).

Exercises III, pp. 78, 79.

1. 23,114 ft.-lb.; 16·51 C.H.U. (29·72 B.Th.U.).

2. In water: 3,151,446 C.H.U. (5,672,603 B.Th.U.); 1,969,654 ft.-tons. **In** steam: 78,767 C.H.U. (141,781 B.Th.U.); 49,229 ft.-tons.

3. $n = \frac{17}{16}$; C = 486·5; 13·76; 4·894; 3·026; 1·871.

4. $a = 0·0134$, $b = 0·002118$; $p = 115$, $w = 0·2570$, $v = 3·89$; $p = 129·5$, $w = 0·2877$, $v = 3·48$.

5. 615·4 C.H.U. (1107·7 B.Th.U.). **6.** 8·76 lb. **7.** 0·8. **8.** 11·92 lb.
9. 614·7 C.H.U. (1106·5 B.Th.U.); 569·3 C.H.U. (1025·6 B.Th.U.) **10.** 1·4417.
11. 0·0707, 0·1987, 0·3121, 0·4140; 2·0656, 1·8882, 1·7577, 1·6554.

12. 0·949. **13.** 0·866, 0·881, 0·896, 0·911, 0·927.
14. 0·523; 0·526. **15.** 17·44 cubic feet; 1·134.
16. 46·12 lb. per sq. inch; 1·319. **17.** 0·952; 0·2149. **18.** 0·98.
19. 0·94. **20.** 0·493. **21.** 0·94. **22.** 0·945.

Exercises IV, pp. 93, 94.

1. 58·3. **2.** Pressures in lb. per sq. inch abs. 100, 42·66, 127·98, 300. Volumes in cubic feet, 1·617, 2·960, 0·987, 0·539. Efficiency, 0·218.
3. 0·5 **4.** 0·291. **5.** (*a*) 0·309; (*b*) 0·854. **6.** Efficiencies, 0·235 and 0·241.

Exercises V, pp. 118–120.

1. (*a*) CO_2 29·1, N 70·9. (*b*) CO_2 21, N 79. (*c*) 294.
2. (*a*) CO_2 19·9, N 72·8, O 7·2. (*b*) CO_2 14, N 79, O 7. (*c*) 441.
3. 3·47 lb.; 15·09 lb.; CO_2 12·0, N 84·3, O 3·6.
4. 11·5 lb.; CO_2 12·7, SO_2 0·1, N 80·2, O 7·0. **6.** 4·86 cubic feet.
7. 0·96; CO_2 14·6, N 81·3, O 4·0. **8.** 10·8 lb.; 6·9 lb **9.** 16 lb.
10. 8·4 per cent.; 18·15 lb.; 13·83 lb. **11.** 25·9 lb.; 11·4 lb.
12. 12060 C.H.U. (21708 B.Th.U.); 22·4 lb.; 15·1 lb.; CO_2 14·1, H_2O 6·1, N 73·5, O 6·3; CO_2 10·1, N 83·8, O 6·1.
13. 7212 C.H.U. (12,982 B.Th.U.); 13·4 lb.; 9·0 lb.; CO_2 14·1, H_2O 8·6, N 71·2, O 6·1; CO_2 10·2, N 83·7, O 6·0.
14. 7885 C.H.U. (14,193 B.Th.U.); 7691 C.H.U. (13,844 B.Th.U.); 14·63 lb.; 14·27 lb.
15. (*a*) 123·5 B.Th.U. (*b*) 0·86 cubic feet (*c*) CO_2 17·1, N 82·9, water vapour condensed. **16.** 11,489 C.H.U. **17.** 891 C.H.U. (1604 B.Th.U.).
18. 1618° C. (2944° F.). **19.** 854° C. (1569° F.); 778° C. (1432° F.).
20. 11,765,657; 16,404,040.

Exercises X, pp. 215, 216.

1. Volume of products—(*a*) 312·8, (*b*) 463·2, (*c*) 613·7 cubic feet. Volume of air—(*a*) 300·9, (*b*) 451·3, (*c*) 601·8 cubic feet.
2. 92·65 feet. **3.** 0·745 inch. **4.** 113·3 feet. **5.** 107 feet.
6. 5·06 feet. **7.** 0·24 per cent.; 11·1. **9.** 11. **10.** 6·6.
11. (*a*) 77·8. (*b*) 0·09 lb. **12.** 6·1; 31·7.

Exercises XI, pp. 228–230.

1. (*a*) 10·2 lb. (*b*) 10·1 lb. **2.** A, 10·3 lb. B, 10·5 lb. C, 11·7 lb.
3. (1) 22·43 lb. (2) 9·15 lb. (3) 10·6 lb. (4) 3·2 lb.
4. 36·8 lb.; 6·7 lb. **5.** 93·4. **6.** 64 per cent. **7.** 73 per cent.; 65·7 per cent.
8. 9·54 lb.; 11·1 lb.; 77 per cent. **9.** 69 per cent.; 9·56 lb.; 9·96 lb.
10. (*a*) 75 per cent. (*b*) 2256 B.Th.U. or 15·3 per cent. of the heat of combustion of the coal.
11. 22·5; 20·1; 0·282; 0·246. **12.** 17·86 lb. **13.** 2·3.
14. (*a*) 1·68. (*b*) 4·11. (*c*) 9·78. (*d*) 7·40

Exercises XIV, pp. 280, 281.

1. 36·5; 36·9; 14·8; 22·9. **2.** 119 lb. per sq. inch: 0·2 of stroke from end of exhaust stroke. **3.** 13·48 inches. **4.** 60·6; 1307 lb.; 21·6 lb.

5.

p_1	120	80	40
H	30·9	18·4	6·0
W	599	409	212

$m = 15·54$. $k = 121$.

6.

p_1	120	80	40
H	31·5	18·7	5·9
W	676	448	212

$m = 18·12$. $k = 107$.

Exercises XV, pp. 303–305.

1. 100. 2. 256·9 and 224·3.
3. (a) 12·95 lb per sq. inch (b) 5·26. (c) 19 6 lb.
4. 23·62 lb. per sq. inch. 5. 21·17 lb. per sq. inch
6. 16·65, 70·5, 1·1, and 4·5, all in cubic feet.
7. H.P., 1·673 lb.; L.P., 1·642 lb 11. 32 lb. per sq. inch; 0 55
12. 64 C.H.U. (115·2 B Th. U.). 13. 19·5 C.H.U. (35 1 B.Th.U.).

Exercises XVI, pp. 315-317.

1. 4041 lb. 2 8517 lb. 3. (b) 0·18. (c) 59·2 lb., 53·1 lb. 4. 0·049
6. 7. 7·64 ft.; 844,437; 241.
8. 5902 lb. 9 58·54 foot-tons. 10. 3943 lb.
11. 1019 lb.; 2 24 per cent.
12. Moments of inertia: 91,377 and 97,974 in lb and ft. units. Weights: 10,153 lb. and 10,886 lb
13. External diameter, 6 feet 3 inches. Section of rim, 7 inches deep and 7·3 inches wide.

Exercises XVII, pp. 338-341.

1 14·09; 1·06; 1·20. 2. 5·9; 11·7, 2·3. 3. 192·3; 0·88 inch.
4. 136·7 revs. per min. 5. 57·8 lb. 6. (a) 1 inch. (b) 217. (c) 219. (d) 218.
7. (a) 50·8. (b) 42·9. (c) 54·4. (d) 46·0. 8. (a) 139·9. (b) 93·6 lb. (c) 11 12 lb.
11. 235; 248; 254; 258. 12. 2 77 inches; 164; 175; 169; 159; 170.
13. 0·606 lb.; 4·04 lb.; 15·5; 2·5. 14. 102·5 revs. per min.
15. (a) 236 (b) 207. 16. 87·7 lb. allowing for gravity effect of balls.
17. 17·8 lb. per inch deflection. 18. 288 and 377 revs. per min.
19. 5·4 lb. 20. 20·9 lb. per inch deflection; $P = 83{\cdot}6\ r - 173{\cdot}9$.
21. 11·6 inches; 300 lb.; 297 revs. per min.
24. 41·3 ft.-lb.; 85 lb.; 6·18 inches.

Exercises XVIII, pp. 383-385.

1. 20,067 lb. or 8·96 tons. 2. 7337 lb. or 3·28 tons.
3. 1806 lb.; 660 lb. 4. 11 inches.
5. 1·65, 5·95, 11·27, 15·95, 18 97; 1·34, 5·00, 10·00, 15·00, 18·66.
6. At cover end, 0·25 inch. At crank end, 0·125 inch.
7. $\theta = 33^\circ$; $\theta_1 = -6^\circ$; $\theta_2 = 119{\cdot}9^\circ$; $\theta_3 = 154{\cdot}8^\circ$; $\theta_4 = -40{\cdot}9^\circ$.
8 $\theta = 38^\circ$; $\theta_1 = -3^\circ$; $\theta_2 = 117^\circ$; $\theta_3 = 144{\cdot}4^\circ$; $\theta_4 = -30{\cdot}5^\circ$.
9. $e = 0{\cdot}23$ inch; $\theta_1 = -6{\cdot}75^\circ$; $\theta_2 = 96{\cdot}75^\circ$; $\theta_3 = 146^\circ$; $\theta_4 = -56^\circ$.
10. $o = 1{\cdot}79$ inches; $i = 0{\cdot}32$ inch; $\theta = 46{\cdot}9^\circ$; $\theta_1 = -3{\cdot}9^\circ$; $\theta_4 = -53{\cdot}9^\circ$.
11. $r = 4{\cdot}07$ inches: $c = 0{\cdot}25$ inch; $\theta = 47{\cdot}5^\circ$; $\theta_3 = 136^\circ$; $\theta_4 = -51^\circ$.
12. $r = 2{\cdot}4$ inches; $\theta = 51{\cdot}3^\circ$; $\theta_1 = -12{\cdot}7^\circ$; $\theta_3 = 140{\cdot}7^\circ$; $\theta_4 = -63{\cdot}4^\circ$.
13. 3 34 inches; 0·67 inch.
14. 4·94 inches; 0·97 inch; 0·16 inch. 1·47 inches; 0·22 inch; 0·01 inch; 1 inch.
15. (1) 60°, (2) 30°, (3) 15·5°. 3·5 inches.
16. Crank side, 1·67 inches; cover side, 1·33 inches; 21·2 per cent.
17. (a) 139°; 160·4°; − 23·4°; − 2·1°.
(b) 91°; 136·6°; − 51°; − 5·4°.
18. (1) $r = 1{\cdot}28$ inches, $\phi = 90^\circ$, $e = 0{\cdot}28$ inch.
(2) $r = 1{\cdot}54$ inches, $\phi = 53{\cdot}6^\circ$, $e = 0{\cdot}24$ inch.
(3) $r = 2{\cdot}125$ inches, $\phi = 31{\cdot}2^\circ$, $e = 0{\cdot}10$ inch.
19. (1) $r = 1$ inch, $\phi = 90^\circ$, $e = 0{\cdot}00$ inch.
(2) $r = 1{\cdot}42$ inches, $\phi = 47{\cdot}1^\circ$, $e = 0{\cdot}04$ inch.
(3) $r = 2{\cdot}19$ inches, $\phi = 32^\circ$, $e = 0{\cdot}16$ inch
20. Cut-off, down stroke, at 63 per cent. of stroke Cut-off, up stroke, at 66 per cent. of stroke. Compression, each stroke, at 85 per cent. of stroke.
21. Cut-off at 85 per cent. of stroke. Compression at 93 per cent. of stroke.
22. 1·29 inches; 42·8°. 23. 0·4 inch. 24. 1369 lb.

Exercises XIX, pp. 395–397.

1. 115·7; 15·7; 86·4 per cent. **2.** 83·6 per cent. **3.** 786; 411·5.
4. 160 revs. per min.; 87·8 per cent.
5. Simple, 1036; compound, 1244. Simple, 0·043; compound, 0·056. Simple, 0·718; compound, 0·723.
6. (*a*): (1) 10·95 lb. per I.H.P. per hour; (2) 111·5 C.H.U. (200·7 B.Th.U.); (3) 0 816. (*b*): (1) 11·78 lb. per I.H.P. per hour; (2) 120 C.H.U (216 B Th U.); (3) 0·798.
7 (*a*) 121·2 C.H.U. (218·1 B.Th.U.). (*b*) 0·194 (*c*) 0·770.

8.

		C.H.U.	B.Th.U.	
Gross heat supplied . . .	per min.	13,396	24,113	100
Heat converted into work . .		1,740	3,132	13
Carried away in exhaust steam		10,720	19,296	80
Unaccounted for		936	1,685	7

9. (1) 434·6. (2) 278·1 (3) 712·7. (4) 23·52 lb. per sq. inch. (5) 12·21 lb. (6) 241. (7) 0·176. (8) 152. (9) 0·631.

Exercises XX (*a*), pp. 409, 410.

1. 0·933; 86 55 lb. per sq. inch; 0·936 lb. per sec. **2.** 0·512 inch.
3. 1449; 1480; 1507; 1520. **4.** 0·515 inch; 1·822 inches.
5. 0·513 inch; 1·816 inches; 0·9833.
6. (*a*) 0·794; 3845 ft. per sec. (*b*) 0·834; 3979 ft. per sec.
7. 0·309 inch; 1·210 inches.
8. (*a*) 6·78 lb. (*b*) 0·256 inch. (*c*) 0·472 inch. (*d*) 98·6° C. (209·5° F.).
9. (*a*) 11·8 lb. (*b*) 0·335 inch. (*c*) 0·643 inch. (*d*) 64·9° C. (148·8° F.).
10. (*a*) 5·69 lb. (*b*) 0·259 inch. (*c*) 1·52 inches. (*d*) 96·5° C (205·7° F.).

Exercises XX (*b*), pp. 430–432.

1. 31° 2′; (*a*) 1381 ft. per sec., 0·844; (*b*) 918 ft. per sec., 0·739.
2. 364; 14·0 lb. **3.** 3643 ft. per sec.; 29° 50′; 1110 ft. per sec.; 63° 54′.
4. 41° 49′. **5.** 9·03 lb. **6.** (1) 3550 ft per sec.; (2) 1300 ft. per sec. (3) 134·6 lb.; (4) 6·75 lb.; (5) 318; (6) 286; (7) 15·7 lb.
8. $\theta_1 = \theta_2 = 23° 14'$; $\theta_3 = \theta_4 = 33° 47'$; $\theta_5 = \theta_6 = 56° 37'$; $\alpha_3 = \beta_2 = 27° 35'$; $\alpha_5 = \beta_4 = 42° 52'$; (*a*) 266·7 lb.; (*b*) 0; (*c*) 106,670 ft.-lb.; (*d*) 0·876.
9. 25° 6′; 30° 41′; 43° 54′; 0·675.
10. (*a*) 0·88; (*b*) 0·86; (*c*) 0·83.
11. (*a*) 27° 7′, 23° 8′; (*b*) 8753 ft.-lb.; (*c*) 0·746.
12. 0·833; 517·1 C.H.U. (930·8 B.Th.U.); 204·7 C.H.U. (368·5 B.Th.U.).
13. (*a*) 182·2 C.H.U. 328 B.Th.U.); (*b*) 138·5 C.H.U. (249·3 B.Th.U.); (*c*) 9·2.
14. 65·7 per cent. **15.** (*a*) 182·4 C.H.U. (328·3 B.Th.U.); (*b*) 93·6 C.H.U. (168·5 B.Th.U.); (*c*) 94·4 C.H.U. (169·9 B.Th.U.); (*d*) 0·67; (*e*) 1·031.

Exercises XX (*c*), pp. 450, 451.

1. 1·31 C.H.U. (2·36 B.Th.U.). **2.** (*a*) 32°; (*b*) 0·68. **3.** 262; 524.
4. 25,858 ft.-lb. **5.** (*a*) 102; (*b*) 330 lb.; (*c*) 1·061 inches; (*d*) 7·767 inches.
6. (*a*) 11·2; (*b*) 2·7. **7.** 94·5 per cent. **8.** 8·33 lb.
9. S = 12K + 7000, where S is the total steam consumption in lb. per hour when the output is K kilowatts: 13·75. **10.** 69·3 per cent.
11. 15·74; 94·5 per cent. **12.** 0·676.
13. 13·25 lb. per Kw. hour; 69·1 per cent.

Exercises XXI, pp. 472, 473.

1. 27·8 inches; 28·2 inches. **2.** 98·6 per cent. **3.** 75 per cent.
4. 52·8 lb. **5.** 35·2 lb. **6.** 0·798. **7.** 0·712 lb. per. sq. inch; 1·410 lb.
8 424. **9.** 414. **10.** Neglecting the effect of the air at the steam inlet (1) 26·16 inches (barometer 29·8 inches); (2) 1·387 lb per sq. inch, and 0·399 lb. per sq. inch; 0·565 lb and 1·227 lb. It is assumed that the air pump deals with the moist air only and not with the condensate.
11. 6·8 lb. **12.** 1, 1·48, and 2·88.

Exercises XXII, p. 514.

1. 597·8 C.H.U.; 543 5 C.H.U. **2** 49·1 C.H.U.; 32·8 C H.U
3. 79·9 C.H.U. (143 8 B.Th.U.); 74 4 C.H.U. (133·8 B.Th.U.).
4. Higher, 74·0 C.H.U. (133·3 B.Th.U.) per c. ft, 1032 C.H.U. (1858 B.Th.U.) per lb. Lower, 70 6 C.H.U (127 B Th.U.) per c ft., 983 C.H.U (1770 B.Th.U.) per lb. Air, 0·9 c. ft. **5.** 3·093 shillings.

Exercises XXIII, pp. 543–545

1. 170 lb. per sq. inch; 436° C. (817° F.).
2. Pressures, in lb. per sq inch abs: At end of compression, 131·7; at end of explosion, 329·3; at end of expansion, 37·5.
Temperatures, in deg. C.: at end of compression, 373·2; at end of explosion, 1343; at end of expansion, 647. M.E.P. 60·8.
3. (1) *a*, 213° C.; (1) *b*, 1447° C.; (1) *c*, 1551° C. (2) *a*, 11·35 C.H.U.; (2), *b*, 1·31 C.H.U.
4. (*a*) 0·327; (*b*) 0·468; (*c*) 0·699. **5.** 13·85; 478·2 lb. per sq. inch.
6. 0 582; 0·704; 0·325. **7.** 0 410.
8. (1) $K_v = 17·61 + 0\ 00688T$; (2) $K_v = 0·161 + 0\ 000{,}0628T$;
(3) $K_p = 0·2377 + 0·000{,}062T - 0\ 000{,}000{,}00725T^2$.
9. 0·2505 C.H.U.; 51·03; 43·28. **10** 1440° C.
11. $K_v = 19·534 + 0·007154t - 0\ 000{,}000{,}466t^2$.
Internal energy $= (19·534 + 0\ 003577t - 0\ 000{,}000{,}155t^2)t$.

Internal Energy in Foot-Pounds.

$t°$ C.	500	1000	1500	2000
By above formula .	10642	22956	36826	52136
" Langen " . .	10605	22930	36975	52740
" Wimperis " . .	10450	22950	37500	54100
" Stodola " . .	10914	3019	36417	50808

12. Internal energy $= (19·56 + 0·00353t)t$ ft.-lb. per c. ft.; 29,323 ft.-lb. per c. ft.
13. Internal energy $= (19·38 + 0·00324t)t$ ft.-lb. per c. ft.; 22,620 ft.-lb. per c. ft. **14.** 82·9 per cent
15. Volumetric efficiency, per cent.: (1) 80·9; (2) 81·1; (3) 81·9; (4) 81·0.
16. 78·7 per cent.; 7·83. **17.** 20·25 C.H.U.; 19·70 C.H.U.
18. 819° C. **19.** 106° C **20** 62·65 lb. per sq. inch; 6·129; 28,544; 1 24.
21. 1852° C.: 1128° C.; 402 lb. per sq. inch; 54·1 lb per sq. inch; 37·1 per cent; 47 per cent.
22 (*a*) 0·1098; (*b*) − 0·0153: (*c*) 0·2073; (*d*) 0·2305.
23. 56584; 25. **24.** 85 lb. per sq. inch.

Exercises XXIV, pp. 567–570.

1. 0·35; 0·703. **2** 16·39; 13·71; 0·278; 0·836.

3. 24·6; 16·34 c. ft.; 0·232. **4**. 563·5.

5. (*a*) 27·81; (*b*) 0·253; (*c*) 15·46 c. ft. **6**. 0·313; 0.631. **7**. 39 per cent.

8. (*a*) 39·02; (*b*) 32·01; (*c*) 82 per cent.; (*d*) 34 per cent.; (*e*) 65·6 per cent. *Heat balance*, per cent.—Heat supplied, 100; heat converted into work, 34; heat lost in jacket water, 29; heat lost in exhaust gases and unaccounted for, 37.

9. 51·9 per cent; 67·8 per cent.; 75·5 per cent

10. G = 13B + 120; 258 c. ft. **12**. 1·04 lb.; 57·2 c ft. **13**. 28·42 C.H U.

14. 130 2 lb. per sq. inch. **15**. Equations to Willans lines.—W = 0·45B + 7·2; W = 0·44B + 5·9; W = 0·46B + 3·7.

16. 109 lb. per sq inch; 27·7 per cent.; 133 lb. per sq. inch; 33·8 per cent.; 73·6 per cent.

17. (*a*) 22·7; (*b*) 0·62 lb.; (*c*) 21·9 per cent.; (*d*) 110 lb per sq. inch.

21. Mechanical efficiency $= \frac{100x}{1\cdot012x + 0\cdot267}$ per cent.

Indicated thermal efficiency $= 47\cdot97 - 7\cdot92x$ per cent.

Brake thermal efficiency a maximum when $x = 1\cdot027$.

22. 0·325 lb.; 0·345 lb.; 41·6 per cent.; 39·2 per cent.

23. 0·690; 0 484; 0·444; 0·446; 0·467.

24. Heat supplied, 100; heat converted into work, 39·6; heat carried away in jacket water, 21·6; heat carried away in exhaust gases, 26·8; heat lost by radiation and unaccounted for, 12·0.

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THE END

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